Noise and Vibration Control for a Decanting Centrifuge

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Abstract

High levels of machine noise result in health issues for those people exposed to the machine for extended periods. Machine noise is becoming a more significant design consideration, often because of legislative requirements.

Decanting centrifuges are machines with a rotating bowl. They have a number of noise sources, the most significant being structural noise, vortex/turbulence noise and siren noise. Due to the size, mass and speed of the rotating bowl, the bowl is the main source of structural noise. The structural noise is produced by all surfaces that are directly coupled to the bowl's bearings. Due to the speed of rotation of the bowl, the turbulence from the various trailing edges generates broad spectrum vortex noise. Siren noise due to air flow through the bowl also generates significant noise especially at the lower harmonics of the bowl rotation frequency.

Strategies to reduce decanter machine noise include:

- Decoupling the surfaces of the from the main bearings of the rotating bowl and hence reduce the amount of structurally transmitted noise.
- Smoothing the surface of the bowl to minimise the edges that produce vortices that are shed and produce vortex/turbulence noise.
- As siren noise is produced due to flow through the rotating bowl, which is integral to the function of the decanter, the exit ports should be designed so that the noise is produced within parts of the sound spectrum that have low noise levels.

1 Introduction

1.1 Machine Noise

Machine noise is a recognised health hazard if exposure is greater than about 80 to 85 dB(A) over an 8 hour day. An exposure level of 85 dB(A) during a working life would result in 90% of people suffering a hearing loss of less than 25 dB when averaged over the frequencies 0.5, 1 and 2 kHz. However, it would be necessary to reduce levels below 80 dB(A) to provide a negligible hearing damage risk for speech. Exposure levels are measured as A-weighted Equivalent Noise Level for an 8-hour exposure ($L_{Aeq,8h}$). In order to limit sound exposure of machine operators there are three possible strategies that are available: reducing the sound power of the machines, limiting exposure of the operator to the noise, or isolating the operator from the noise of the machine [1]. The focus of this report is on the reduction of sound power of the machine. The machine used for this study is a G-Tech Bellmor 1456 decanting centrifuge.

1.2 Mechanisms of Noise Production

Noise is unwanted sound. Sound is due to pressure fluctuations, normally in air, that are detected by the ear. The range of frequencies that are detectable by human hearing range from 20 Hz to 20,000 Hz. Machines can produce noise in a variety of ways. Noise production can be categorised into one of the following mechanisms [2]:

- Vortex Noise – Vortex noise is from the vortices that are shed periodically from wakes of trailing edges of moving surfaces. An example on the decanter is the bolt heads that hold the bowl segments together and the wear plates at the solid discharge ports.
- Siren Noise – Siren noise is from periodic air compression, where air is trapped in a semi-enclosed region which is compressed and emits a series of air explosions. An example in the decanter is the air flow out of the bowl ports and into an air space which fluctuates in size due to the rotation of the bowl.
- Acceleration Noise – Acceleration due to impacts cause surfaces to vibrate. This was not a mechanism present during testing of the decanter but would be during normal running due to solids and liquids flowing out of the rotating bowl and impacting the hopper.

- Vibrational Noise – Vibrational noise comes from imbalances within the machine that results in vibrations of the machine's surfaces. An example is imbalance of the decanter's rotating bowl causing vibrations within the base, hopper and gearbox guard.
- Frictional Noise – Frictional noise is from one surface moving over another surface. An example of this is v-belts of the decanter moving against the drive pulley.

According to Lucas et al. (1997), in assessing machine noise, factors which should be considered are structure-borne noise, air-borne noise and fluid-induced noise within fans and impellers and Pérez, (2009) discusses siren noise and the relationship between the pressure fluctuations and surrounding geometry [3, 4]. Noise and vibration control strategies include damping treatments and isolation of structures to reduce the transmission of energy to the surroundings [5, 6]. Another method of vibration control is to use mass as a dynamic load for vibration absorption [7].

Vibration analysis along with sound measurements assist in determining the likelihood of a structural noise source being a significant cause of the noise at specific frequencies; the relevant machine component can be separated blindly out of a set of acoustical or vibration measurements [8]. Other researchers, such as Vijayraghavan, P. (1999), also provide categorisation of machine noise sources and possible attenuation methodologies [9, 10].

This work focuses on vibrational/structural noise, vortex/turbulence noise and siren noise. These areas cover the main sources of noise and significant reductions in these noise levels due to these sources can potentially be achieved with relatively little cost.

The decanter used for testing was new. It was assumed that the bowl was in optimal condition and balanced. As the rotating bowl was the main source of machine vibration any improvements in the balancing of the bowl would result in a reduction in sound power from the decanter. The main bearings at each end of the bowl would also be sources of noise. Due to the relatively low noise levels from the bearings, compared to other sources, the bearing noise was not investigated.

There were two electric motors on the decanter. The main drive motor was used to rotate the bowl and the back drive motor was used to rotate the auger, via a planetary gearbox.

The power from the motors were transferred to the decanter via v-belts. These parts of the decanter were not modified.

The decanter was tested in a laboratory space within the Department of Mechanical Engineering at the University of Canterbury. The space was reverberant in nature and was consequently acoustically treated. The laboratory space was used by other project and the acoustical characteristics were likely to change between tests. For this reason the sound power of the decanter was assessed using the sound intensity method. Research has shown that this was an effective method of assessing the sound power of a machine in these conditions [11, 12].

2 Decanter Running Conditions

2.1 Introduction

The noise emissions from a continuous discharge decanting centrifuge – hereafter referred to as a decanter, were considered to be best assessed using sound intensity techniques. The parameters for using this technique are established and reported here and can be used in establishing a base case against which noise levels due to modifications to the decanter will be compared.

The particular characteristics of the decanter that need to be addressed when there are many sound sources include the air flow around the decanter, establishment of the scanned areas, and the required acoustic environment in which measurements are to be made. The quality of the sound intensity measurements is also addressed.

The decanter was a G-Tech Bellmor 1456 (serial no.: GDI-18-KE-3767-1) running at normal operating speeds. The main motor control was set at 48.7 Hz corresponding to a bowl speed of 3248 RPM. The back drive motor control which drives the auger was set at 47.8 Hz.

2.2 Equipment Setup

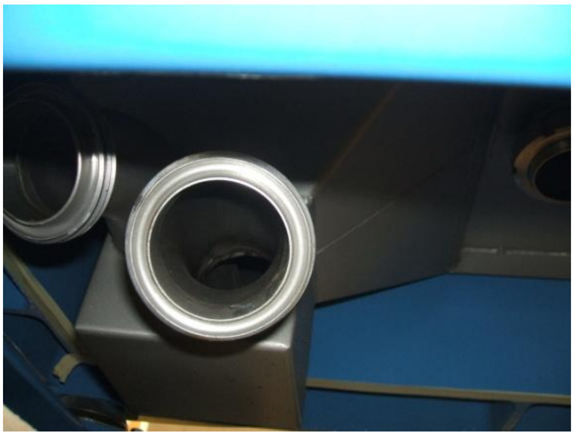
To quantify the change in sound transmitted from the decanter due to modifications, the noise produced by the decanter had to be determined accurately. Given the comprehensive noise measurement equipment available and the location of the decanter for testing, sound intensity measurement was selected as the most appropriate technique. The transmitted sound intensity was measured using a Brüel and Kjær Type 2260 Sound Investigator (serial no.: 1894145) with Brüel and Kjær BZ7205 Sound Intensity software and a Brüel and Kjær Type 3595 Sound Intensity Probe Kit (serial no.: 2087707). The sound intensity probe kit was calibrated using a Brüel & Kjær Type 4231 Acoustic Calibrator (serial no.: 2309461) and a Brüel & Kjær Type DP0888 Adaptor.

As the decanter produces a significant amount of air movement when running, from the motors and exit ports of the rotating assembly, the effects of the use of a windscreen over the intensity probe was evaluated. Measurements were compared in an airflow and in a situation with no airflow.

2.3 Test 1: Sound Intensity within an Airflow

Using the exit port (see Figure 1(a)) of the decanter as a noise and wind source the intensity was measured for a period of 30 seconds with the intensity probe 200mm from the port. This was repeated with and without a windscreen over the intensity probe. The bowl of the decanter rotates at 54 Hz and peaks in intensity were consequently expected in the 50 Hz and 100 Hz one-third octave bands. The results are shown in Figure 2. The results show that below the 100 Hz one-third octave band the intensity measured without the windscreen is significantly higher than with the windscreen; the overall intensity measured without the windscreen was 7 dB higher than with the windscreen. The results with the windscreen follow the predicted trend in the lower frequency bands, with local peaks in intensity at the 50 Hz and 100 Hz one-third octave bands. The use of a windscreen lowers the measured noise level when measurements are made around the ports and this is attributed to the reduction in turbulent pressure fluctuations on the microphone diaphragms due to the airflow. This turbulence was significant enough to affect the total intensity measured.

a Exit port of decanter



b Exit port of decanter

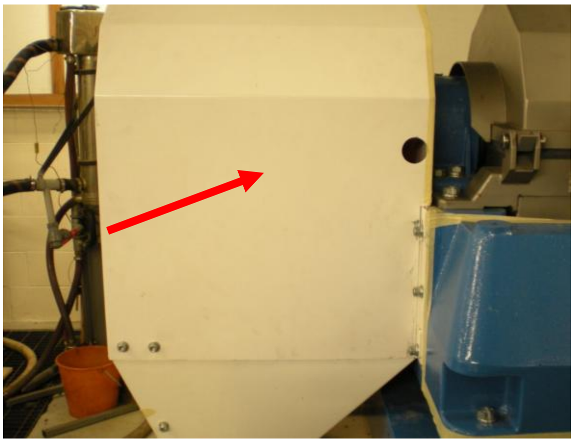


Figure 1 Exit port of decanter

2.4 Sound Intensity without Airflow

Using the centre of the front side of the decanter gearbox guard as a sound source, see Figure 1 (b), the sound intensity was measured for a period of 30 seconds with the microphones 200mm from the gearbox guard. The results are shown in Figure 3. The difference in total intensity measurements was 0.6 dB. Apart from the 4000 Hz one-third octave band, there was less than 2 dB difference in intensity for the octave bands of the two measurements. There was a 6.9 dB difference for the 4000 Hz one-third octave band when using the windscreen and this was probably due to the measurement without the windscreen not detecting a peak in intensity in this one-third octave band. The conclusion from this test was that the use of a windscreen when measuring intensity gave comparable results to when not using a windscreen.

The above two tests on the effects of using a windscreen suggest that using a windscreen will (a) minimise the effects of air flow when measuring intensity and (b) not affect results when measuring away from an air flow. Therefore all subsequent intensity measurements can be carried out using a windscreen. To ensure tests results are not affected by air flows the two fluid exit ports, shown in Figure 1 (a), and the hopper lid exit port were blocked off and the solids discharge chute was also covered.

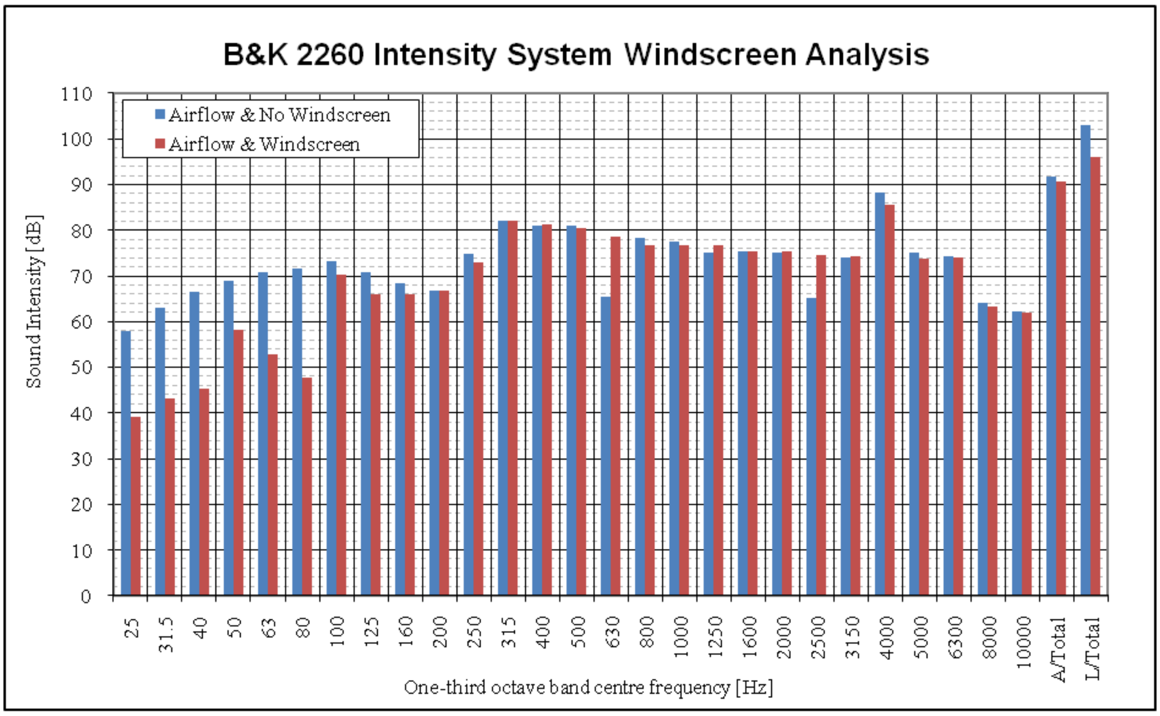


Figure 2 Intensity measurement in an airflow

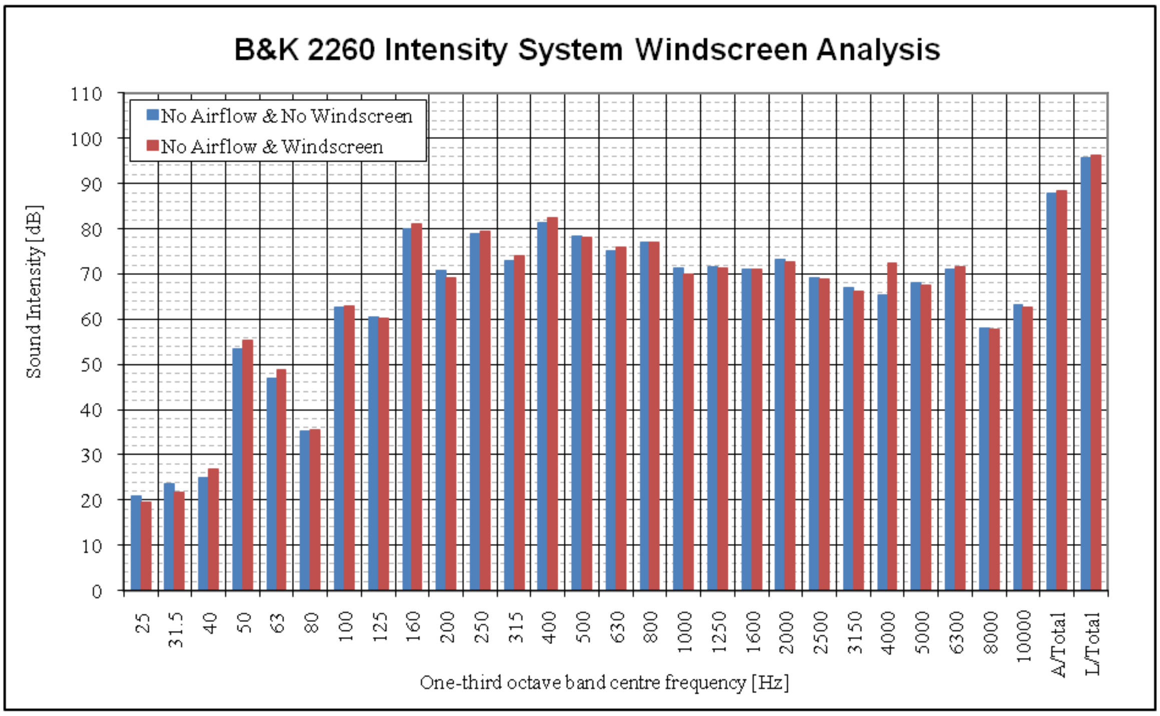


Figure 3 Intensity measurement with no airflow

2.5 Evaluation of the Testing Room

The decanter was tested within room C154 of the Mechanical Engineering laboratory wing of the University of Canterbury. The acoustic environment of this laboratory was assessed to determine if it was likely to affect results of acoustic testing. The reverberation time was measured using a Brüel and Kjær Type 2260 Sound Investigator (serial no.: 2320962) with Brüel and Kjær BZ7204 Building Acoustics software and a Brüel and Kjær Type 4189 Microphone (serial no.: 2607720). The microphone was calibrated using a Brüel & Kjær Type 4231 Acoustic Calibrator (serial no.: 2309461) and JBL EON Power10 Loudspeaker (serial no.: J205-036569).

The testing was carried out using four speaker and microphone positions and twelve measurements were taken. Due to the laboratory having a large amount of steel ducting (see Figures 4 to 6), sound absorption material was added around the surfaces closest to the decanter. The material used was a black 22 mm thick polyurethane foam and a white 30 mm thick polyester board.

Figure 4 Sound absorption behind decanter

Figure 5 Sound absorption in front of decanter

Figure 6 Other area of room C154

The results of the reverberation time measurements are shown in Figure 7. The results show higher reverberation times in the 63 Hz and 80 Hz one-third octave bands. This is most likely due to the large amount of duct surfaces. Frequencies within these two bands are difficult to attenuate due to their long wavelengths. Measuring intensity in these two, one-third octave bands will also be more difficult as measured intensity levels will decrease as the room becomes more reverberant.

The decanter has been placed on NDF sheets. At the main drive end, the decanter sits on a concrete floor. At the gearbox end, the decanter sits on a metal mesh floor which covers a work pit. The position of the decanter on the two flooring surfaces is shown in Figure 8. The mesh floor was not ridged and the natural frequency was determined by placing an accelerometer on top of the left rear leg and exciting the floor, the results are shown in Figure 9. The natural frequencies of the mesh floor were 27, 50, 54, and 65 Hz. The 54 Hz natural frequency was the same as the running speed of the rotating bowl. This indicates that the gearbox end of the decanter will support higher vibration levels at the running speed of the rotating bowl than the main drive end of the decanter.

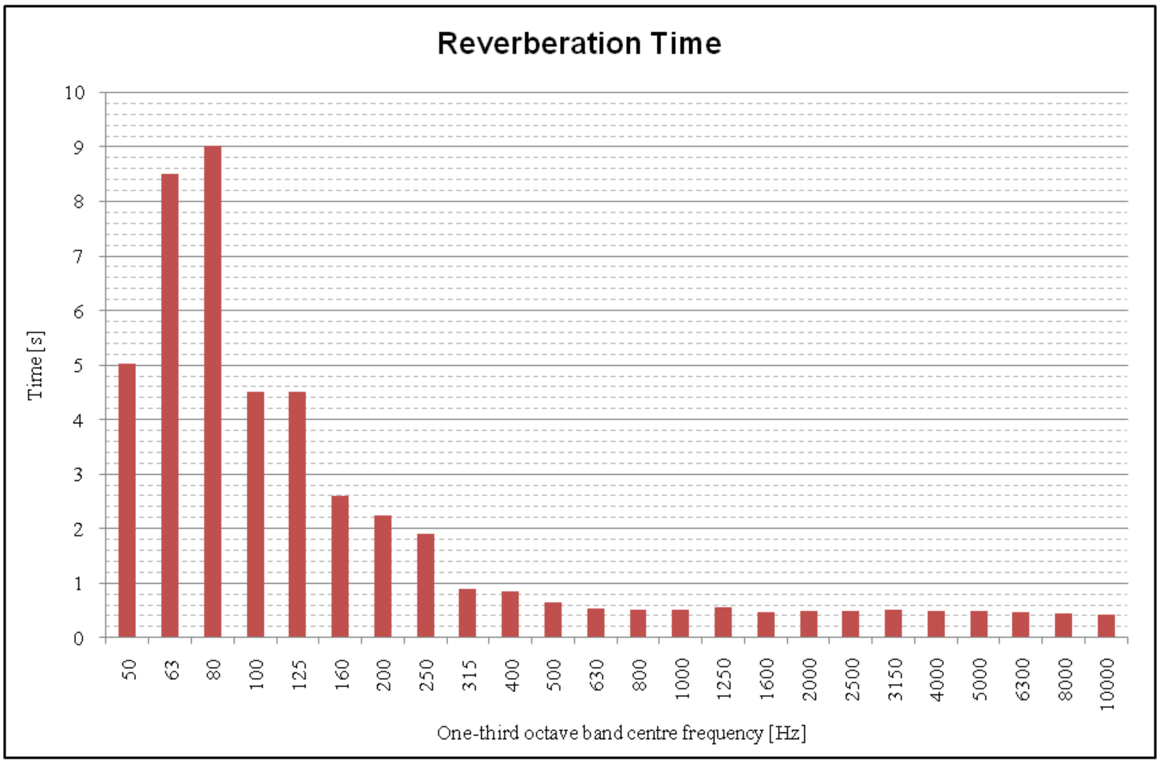


Figure 7 Reverberation time of room C154

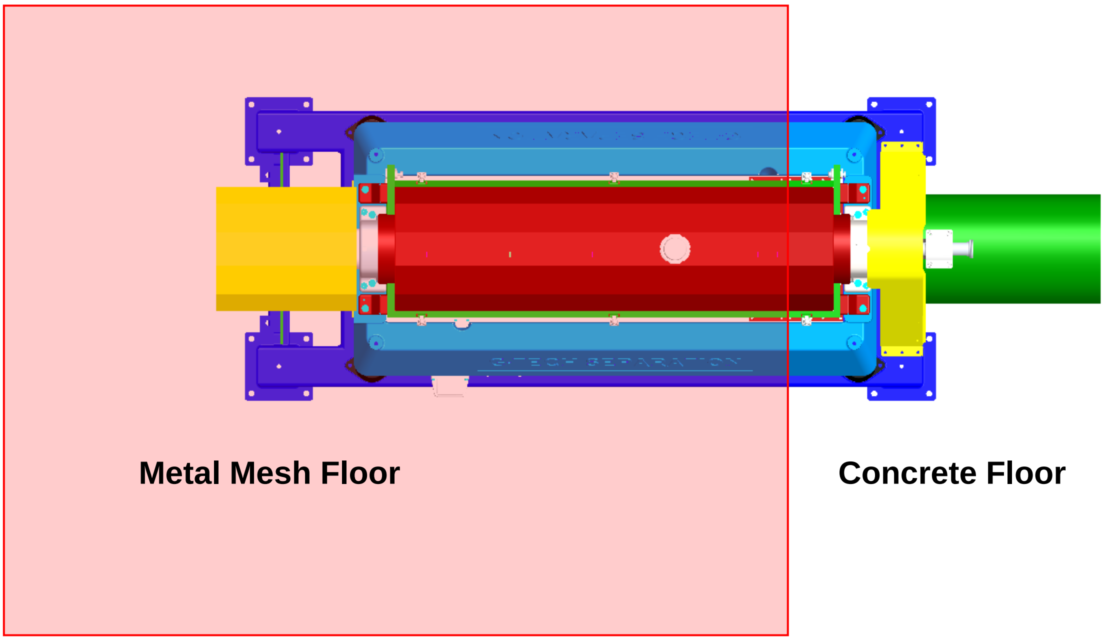


Figure 8 Position of decanter and flooring

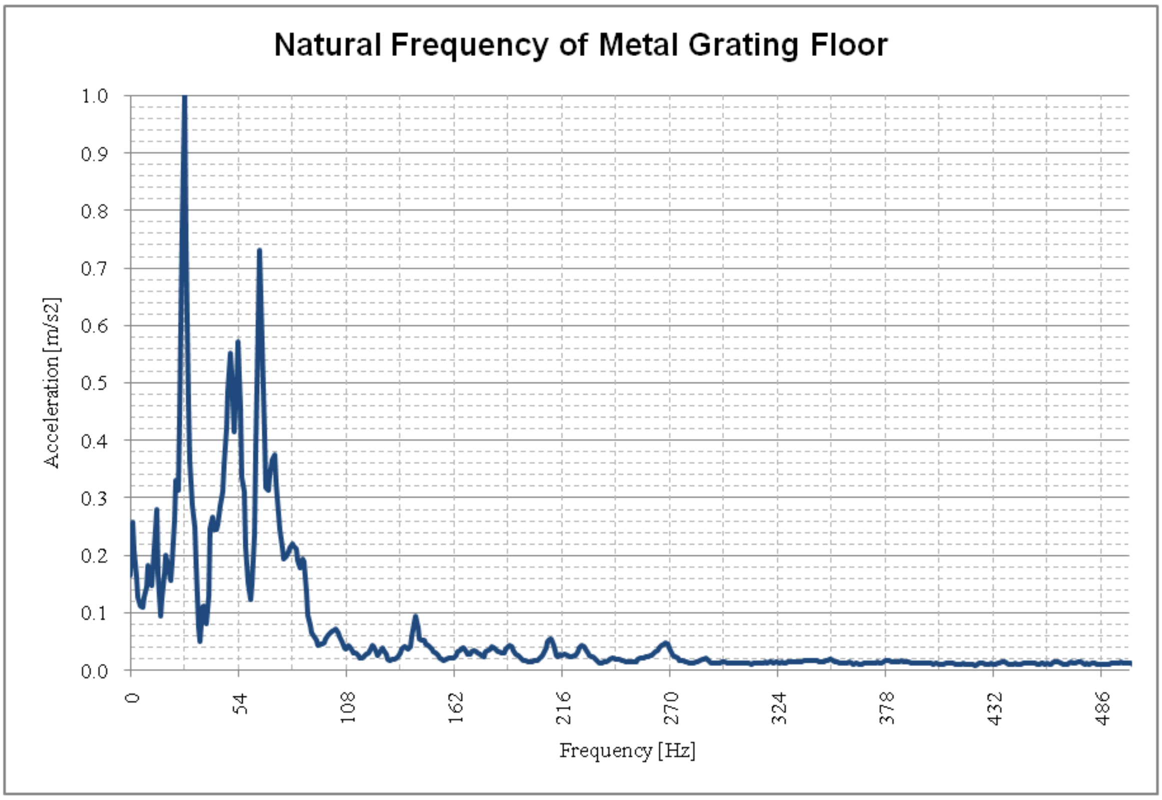


Figure 9 Natural frequency of metal grating floor under the decanter legs at the gearbox end

2.6 The Effect of Scan Area on Measurement Accuracy

The decanter was scanned using the sweep method. The sweep method involves enclosing the machine in a number of scan surfaces. The intensity probe is then swept over each surface to determine the average sound intensity of each surface. In Figures 10 and 11, the areas of the surfaces for the generalised scans of the decanter are shown. The decanter was initially scanned using one scan per side, as indicated by the red box enclosing the decanter. The scan was then repeated with two scans per side, the area being divided into two as shown by the black line. For the last scan, the areas were divided again into resulting quarters, as shown by the white line, and four scans per side were undertaken. The top was divided in a similar manner as the front. The horizontal division was based on the top of the sub-frame. The vertical division was based on the central lock for the hopper for the front, back and top sides and the centre of the area for the left and right sides.

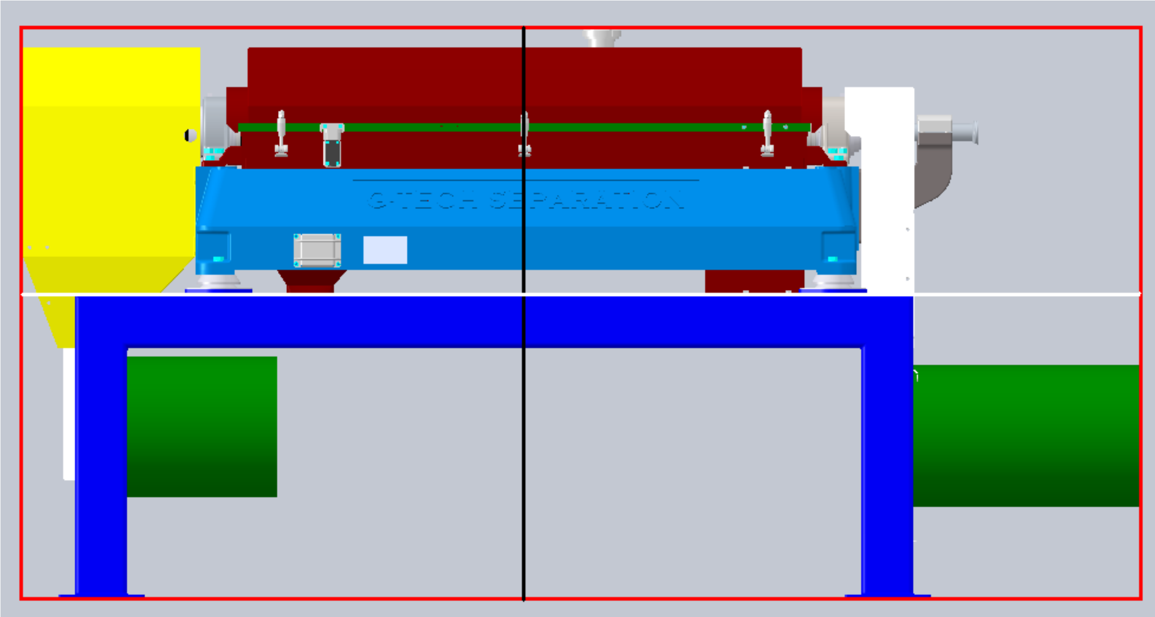


Figure 10 Front of the decanter showing layout of areas for generalized scans.

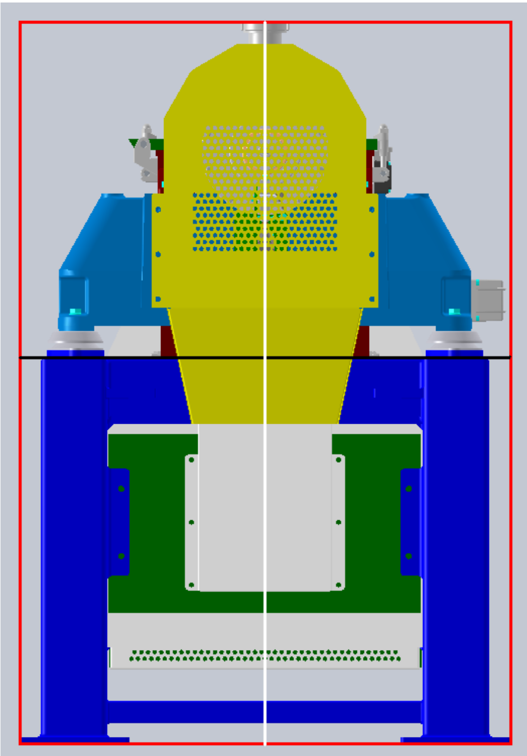


Figure 11 Left end of the decanter showing layout of areas for generalized scans.

The results of the intensity scans are shown in Figures 12 and 13. The results show that the size of the scan areas had no significant effect on the overall measurements. From Figure 12 the sound power measured through the left hand side increased as the area was divided into smaller scan areas. As the decanter has the bowl and auger rotating at slightly different speeds the resulting noise from the decanter was not constant but cyclic. This was particularly evident in the sound from the gearbox guard. To ensure that a representative average intensity was measured the area had to be divided up into regions of similar sound intensity and scanned at a slower rate. The slower scan rate allows more measurements to be taken for calculation of the average intensity for that area. The boundaries of the areas should ideally be away from high intensity regions so that these regions are not over represented in the results. The areas also need to be manageable and one square metre is about the maximum size that can be scanned by the operator in a stationary position. The areas should also be divided up based on the position of the main components of the decanter. This would allow the identification of the sound power for each component.

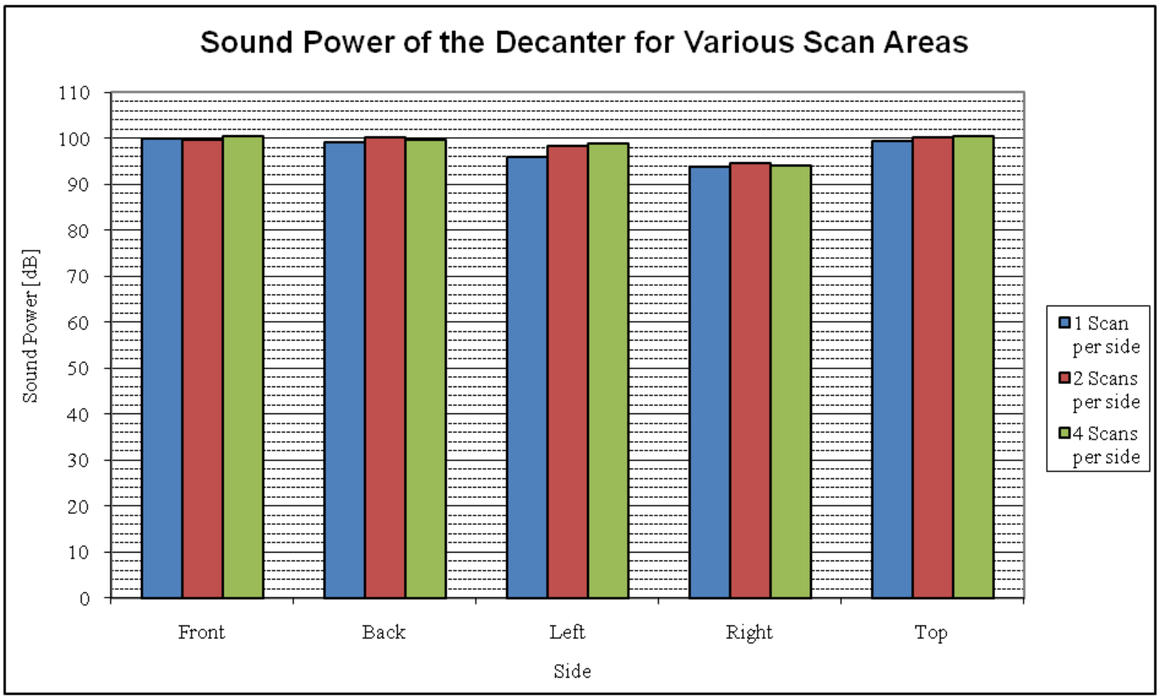


Figure 12 Sound power of each side for various scan areas

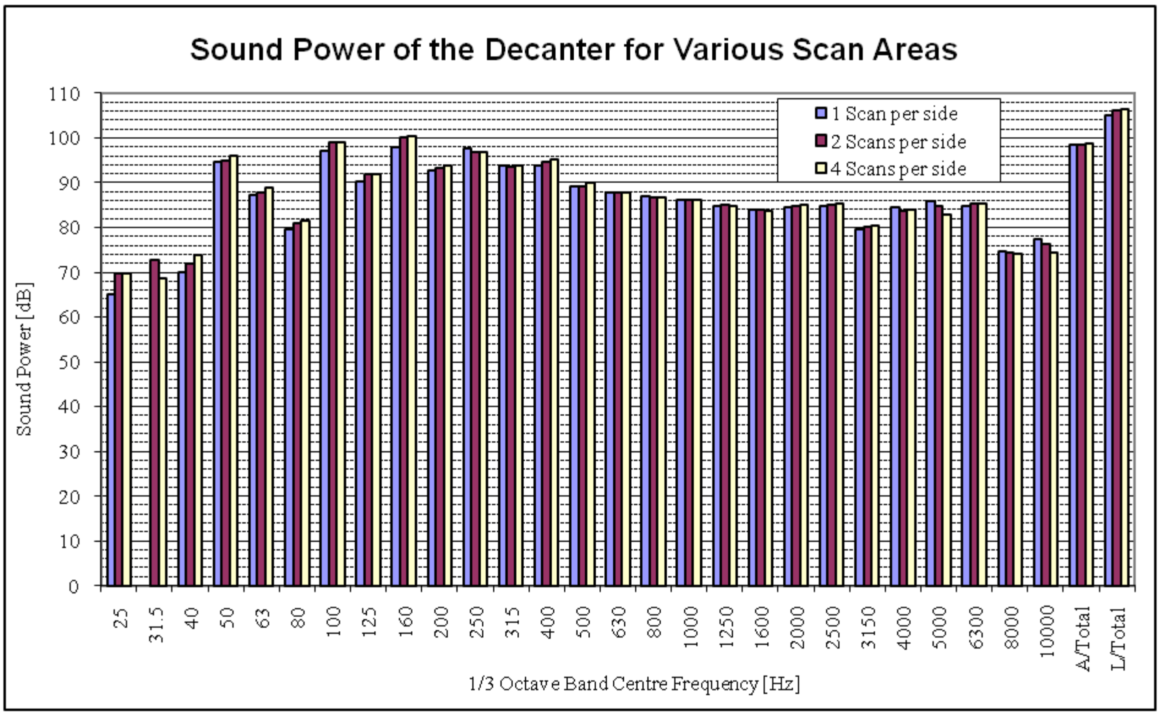


Figure 13 Sound power by number of scans

2.7 Division of Regions for Intensity Scans

The division of the areas of the sides for subsequent testing are shown in Figures 14 to 17. A virtual box was created that enclosed the decanter and the surfaces of this box were divided into sub regions approximately one square metre each for scanning. The front and back surfaces were divided into eight regions as shown in Figures 14 and 15. The horizontal division was along the top of the sub frame and the vertical divisions were at each end of the base and at the centre lock for the hopper. The top surface, Figure 16, was divided into four with divisions based at each end of the base and at the centre lock for the hopper. The left surface, Figure 17 (a), was divided into four regions. The horizontal division at the top of the sub frame and the vertical divisions at the edge of the gearbox guard. The right surface, Figure 17 (a), was divided into two regions with the horizontal division based at the top of the sub frame.

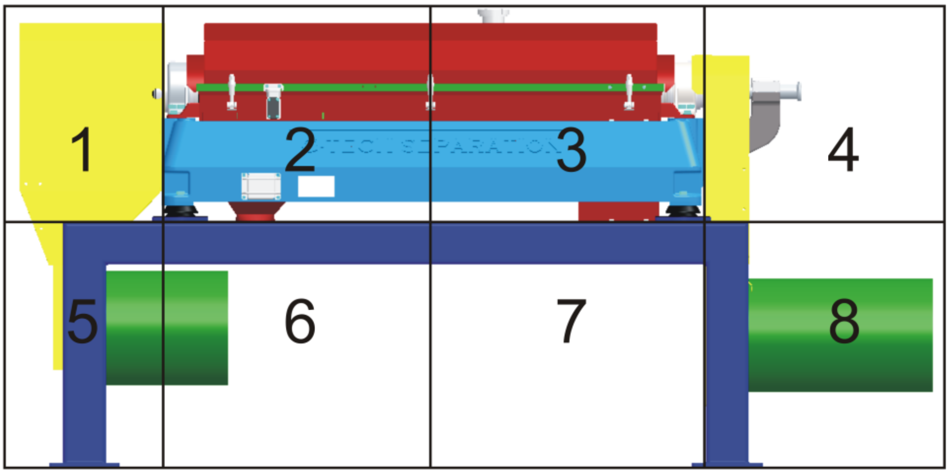


Figure 14 Front scan areas

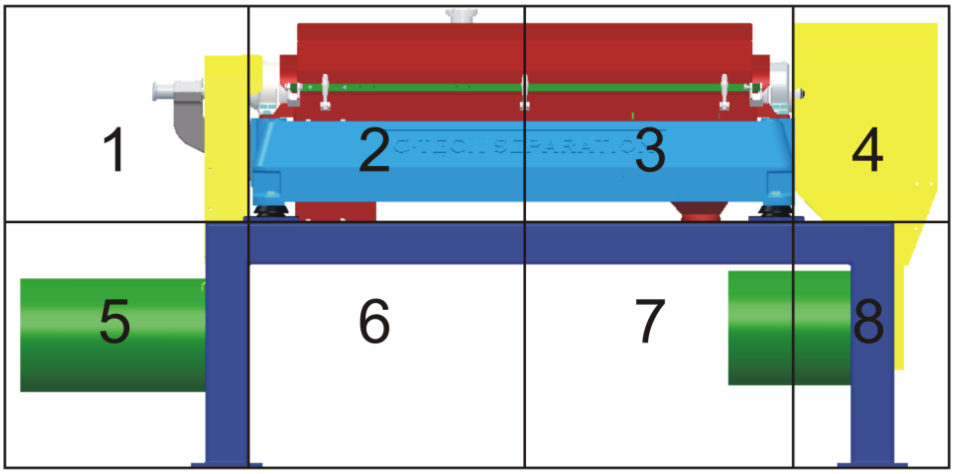


Figure 15 Rear scan areas

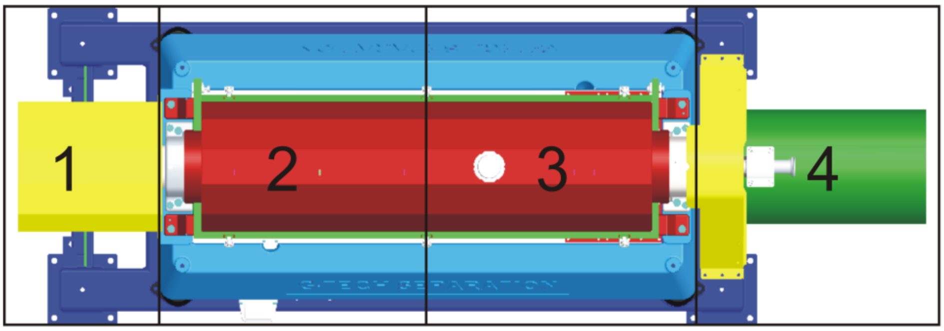


Figure 16 Top scan areas

a Left scan areas **b** Right scan areas

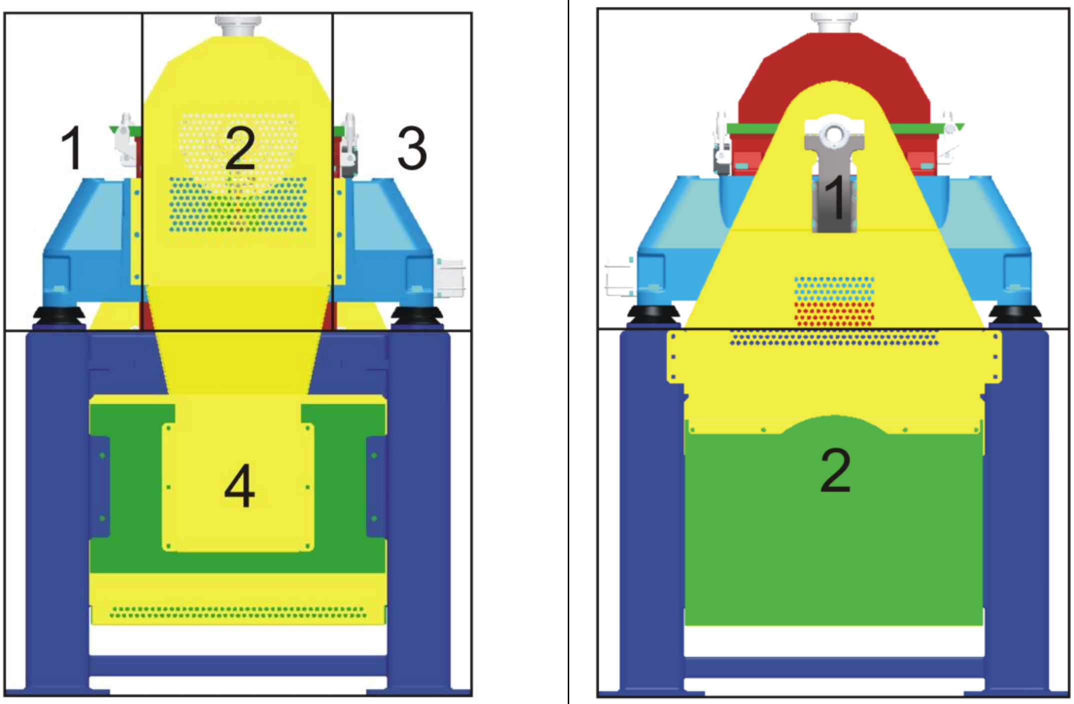


Figure 17 (a-b) Left and right scan areas

2.8 Quality of Sound Intensity Measurements

The pressure-intensity index δ_{pI}, is the most widely used sound field indicator for sound intensity measurements. This quantity provides some general indication of the state of the sound field at the point of measurement. Where δ_{pI} differs markedly from zero there is a specific physical reason such as [14]:

- The probe axis is nearly perpendicular to the direction of the mean intensity vector.
- The field is partially reactive (e.g. in the hydrodynamic near field of the source, or in a multiply reflected coherent field).
- The field is effectively ‘diffuse’ (e.g. not close enough to the source).
- The field is produced by two or more sources which generate oppositely directed intensity vectors of similar magnitude at the point concerned, which almost cancel each other.

Scans of the front of the decanter were undertaken under various operating conditions. The objective was to determine what conditions were required to obtain ‘quality’ data. The scanned area is shown in Figure 14. The pressure-intensity index was used to determine the quality of the measured results. The results from the testing are shown in Figures 18 to 24. The black lines on the pressure-intensity plots indicate the maximum level for the measurement to be considered good ‘quality’.

The first test was conducted with the standard amount of absorption placed within the room, as shown in Figures 4 and 5. The results of the measurements are shown in Figure 18. The results show that area 4 and 8 have a relatively high δ_{pI} for all frequencies. This is due to the low noise radiating from the decanter parts within these regions compared to the noise levels generated within the adjacent regions. Overall the results can be considered good quality for the 200 Hz one-third octave band and above. The lower frequency bands cannot be classified as good quality using the δ_{pI} measurements.

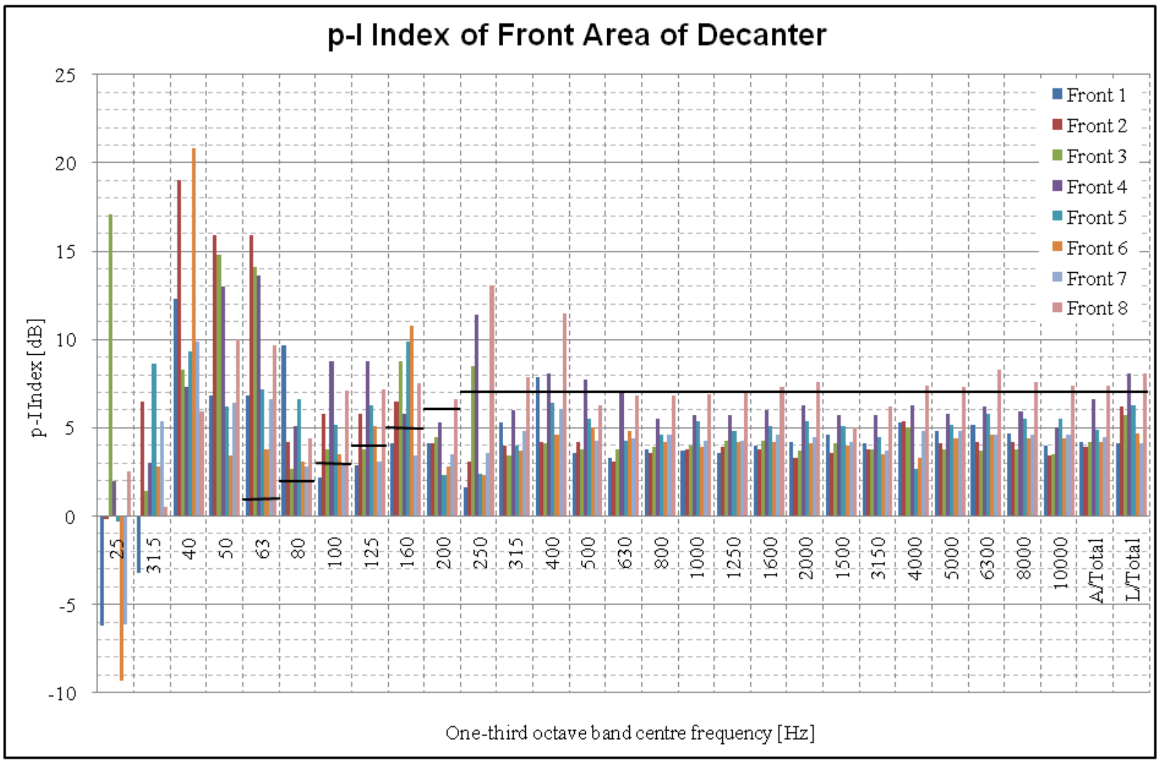


Figure 18 P-I index for the first test

The second test was conducted with sound absorption material added to the room as shown in Figure 18. The added absorption consisted of a 100 mm thick polyurethane foam sheet 1.2 x 2.4m (A) and 4 x 50 mm thick polyurethane foam sheets 1.2 x 1.4m (B). The results of the measurements are shown in Figure 20. The results show a slight increase in quality over all the frequencies but the measurements below the 200 Hz one-third octave band can still not to be classified as good ‘quality’.

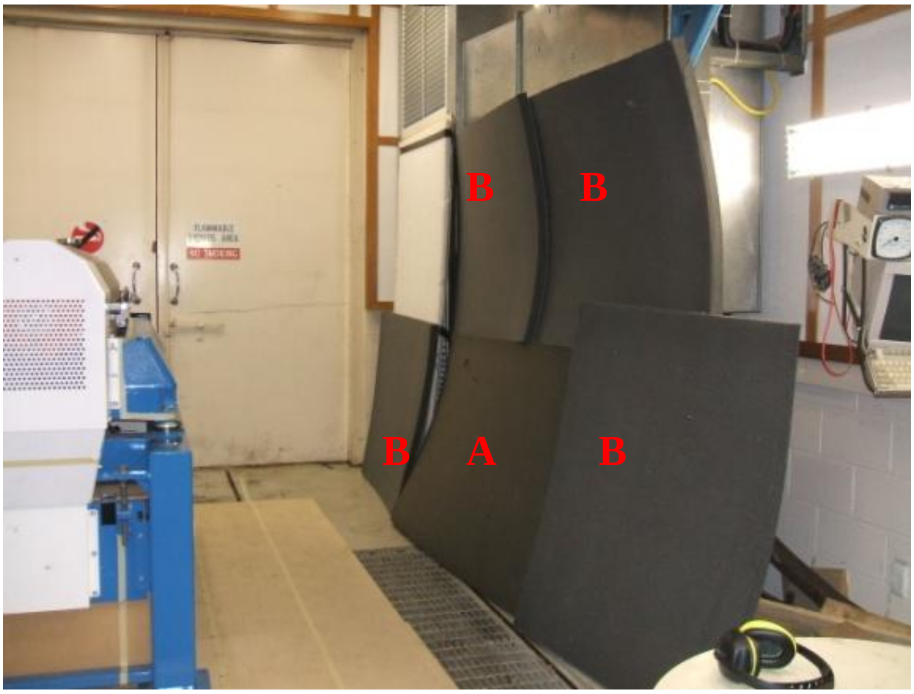


Figure 19 Additional Sound Absorption

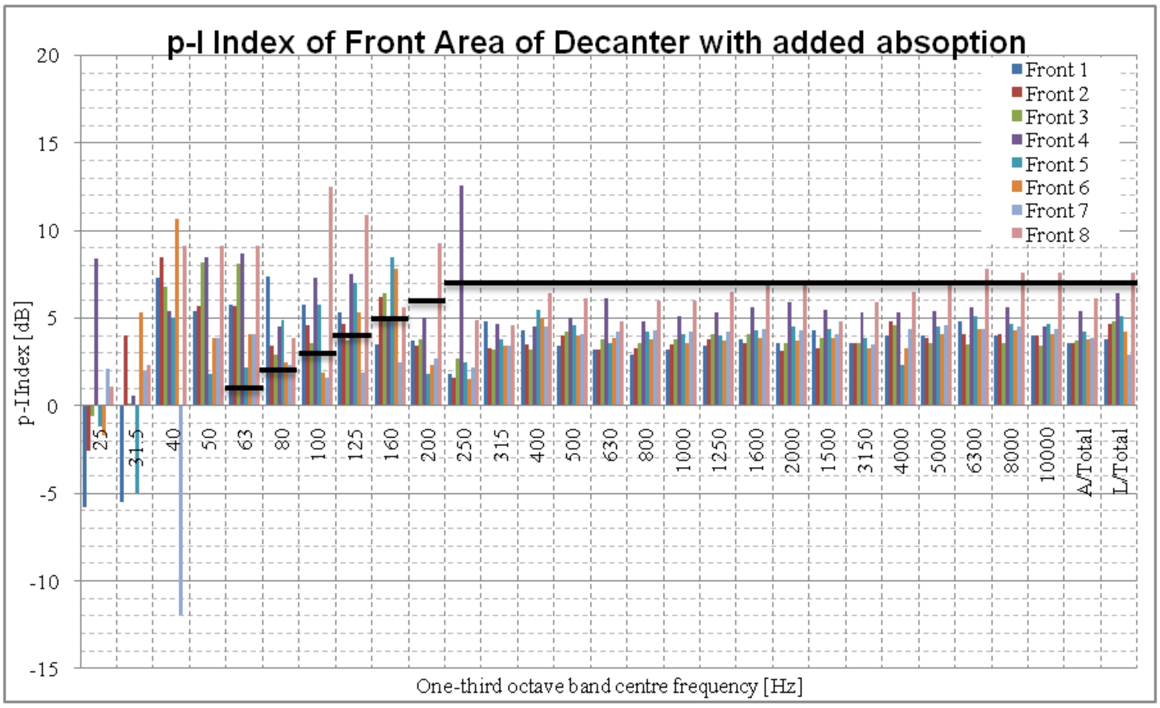


Figure 20 P-I index for the second test

The third test was conducted with the same sound absorption as used for the second test, with additional absorption material placed behind the decanter as shown in Figure 21. This additional absorption consisted of 2 x 50 mm thick and 4 x 25 mm thick polyurethane foam sheets 1.2 x 1.4m. The results of the measurements are shown in Figure 22. The results are very similar to the results of the second test.

Figure 21 Additional absorption added behind the decanter

The fourth test was conducted with the same sound absorption as used for the second test, but using a 50 mm spacer in the intensity probe instead of a 12.5 mm spacer. The results of the measurements are shown in Figure 23. The results show good quality data for 80 Hz – 4000 Hz one-third octave bands but the remaining one-third octave bands still do not have good 'quality' measurements.

The sound powers for the four tests are compared in Figure 24. The total sound power measured in all four tests was within 0.6 dB for all tests. From the four tests, the sound power for the 80 to 2000 Hz one-third octave bands were all within 2 dB. Above the 2000 Hz one-third octave band the measurements from test four, using the 50 mm spacer, were significantly lower than for the first three tests. The overall sound power levels are all within 1dB of each other.

Test four shows good quality results for 80 to 200 Hz frequency bands. As the results for test one through three are very similar to the result of test four, they therefore can also be considered of good 'quality', even though their δ_{pI} measurements do not indicate good 'quality'.

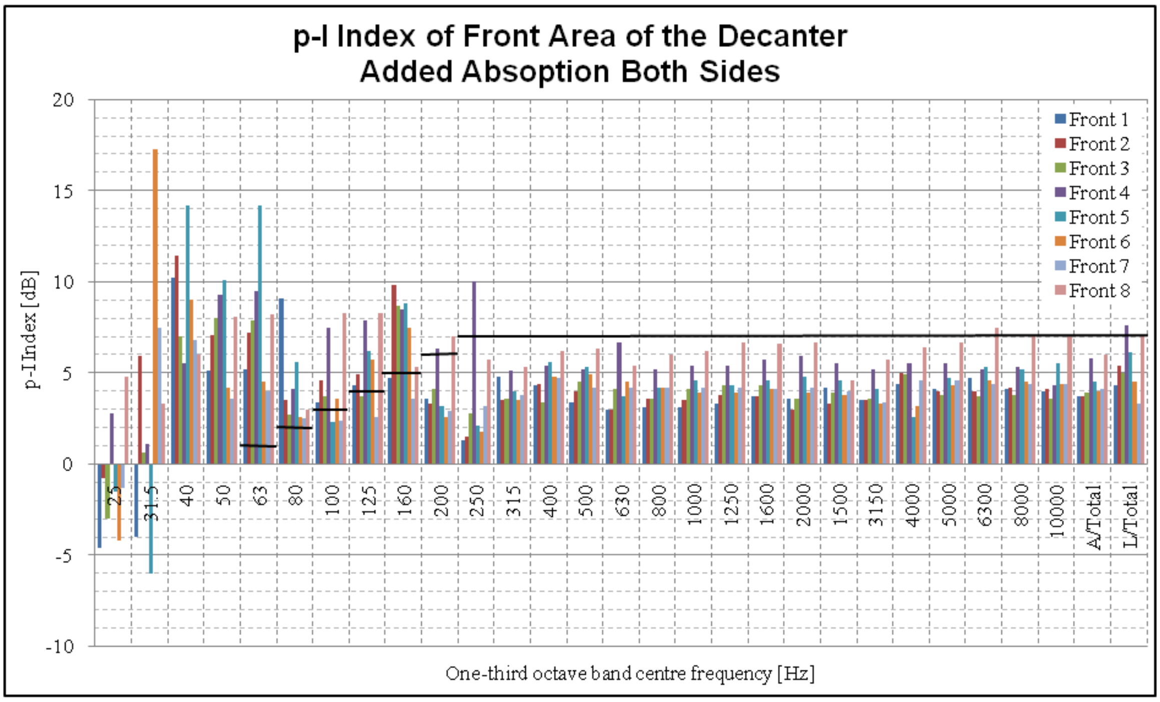


Figure 22 P-I index for the third test

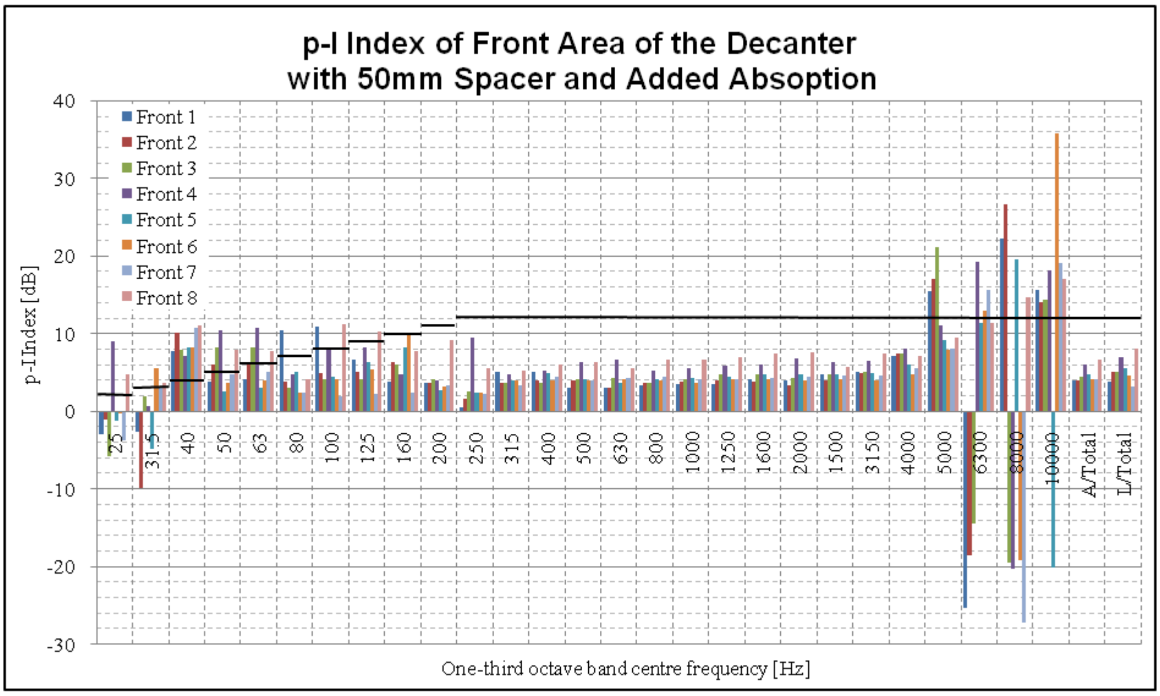


Figure 23 P-I index for the fourth test

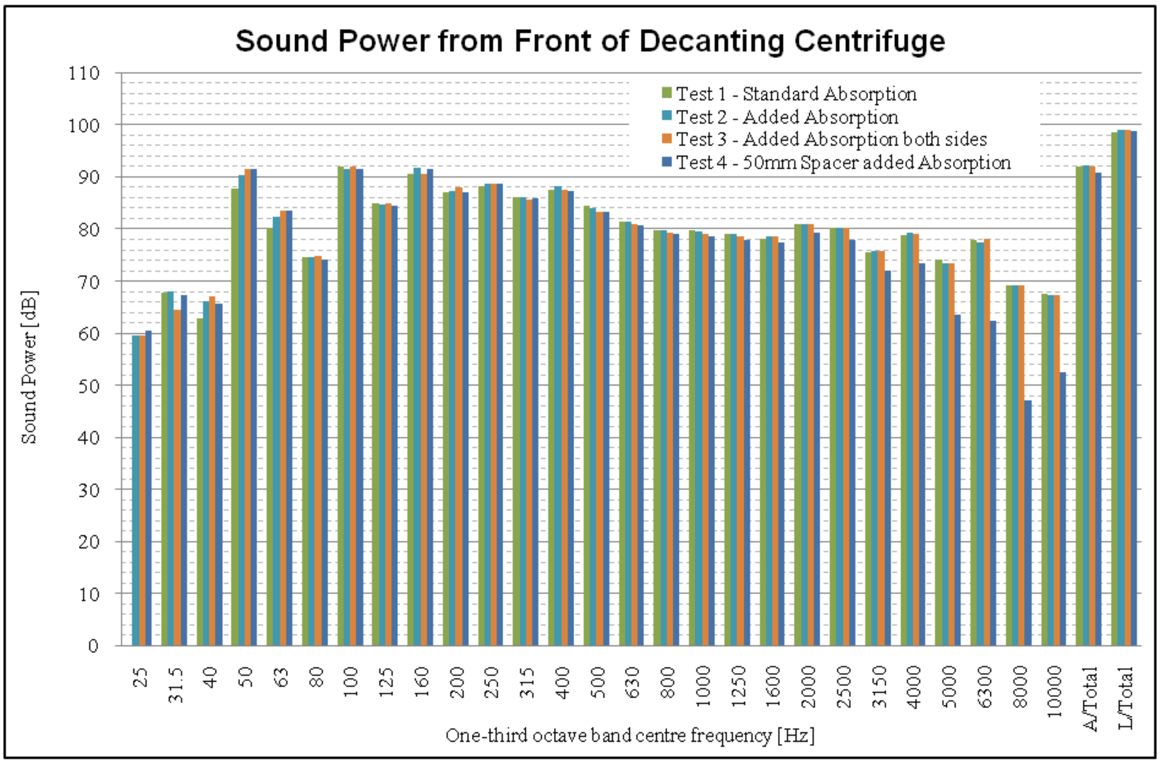


Figure 24 Sound power of the decanter

The results for all four test show frequent negative δ_{pI} measurements in the 25 and 31.5 Hz one-third octave bands. This also relates to intensity measurements that are negative or positive for the same area. As the sound power in these two frequency bands is relatively low in comparison to the remaining one-third octave bands, their fluctuations do not affect the overall sound power measurement. The 25 and 31.5 Hz frequency bands will not be presented in subsequent analysis as the changes may not represent actual changes in the emitted. The 40 Hz one-third octave band will be presented as an indication of the sound power at frequencies below the fundamental vibration frequency of the decanter, 54 Hz, but no conclusions should be drawn from changes in measurements for this frequency band.

The 50 and 63 Hz one-third octave bands need to be presented, as the fundamental vibration frequency of the decanter is 54 Hz due to the rotation speed of the bowl. The tests show that with the high reverberation times in the lower frequency bands it is not possible to generate good quality measurements without significant modifications to the testing room. Therefore any subtle changes in sound emissions in these two frequency bands cannot be used to draw conclusions as to the effects of physical modifications to the decanter.

2.9 Acceleration Measurements

Acceleration measurements points on the base, gearbox guard and hopper are taken to assess the vibration levels within the decanter. These measurements will be compared with measurements of subsequent decanter modification to assess the effect of the modification on the vibration levels within the decanter. The location of the measurement points are illustrated in Figures 25 to 27. The measurement points are either on the base, gearbox guard or hopper and are measuring accelerations in the vertical, lateral or other direction. Table 1 shows how the measurement points were categorised. The acceleration measurements are combined by taking the 'root of the mean of the squares' (RMS) of the various individual measurements. The labels along the frequency axis will the harmonic frequencies of the rotating bowl.

The acceleration measurements were made using a Brüel & Kjær PULSE analyzer and a 352C33 High Sensitivity ICP Accelerometer (serial no.: 88139). Measurements were made at 1 Hz resolution over the frequency range of 1 to 1600 Hz. Using a Hanning time weighting window, an average of 177 measurements (45 seconds) was used as the vibrations were cyclic in nature due to the different rotation speeds of the bowl and auger.

Table 1 Grouping of measurement points

Component	Base	Gearbox Guard	Hopper
Vertical Measurements	1, 20, 30, 31, 32, 33, 34, 73, 74	22, 23, 24	8, 9, 10, 11, 12, 13, 14, 19
Lateral Measurements	35, 36, 37, 38, 39, 71, 72	25, 26, 27	15, 16, 17, 18, 40, 41, 42, 43, 44
Additional Measurements		28, 29, 61, 63, 64	65, 66, 67, 68

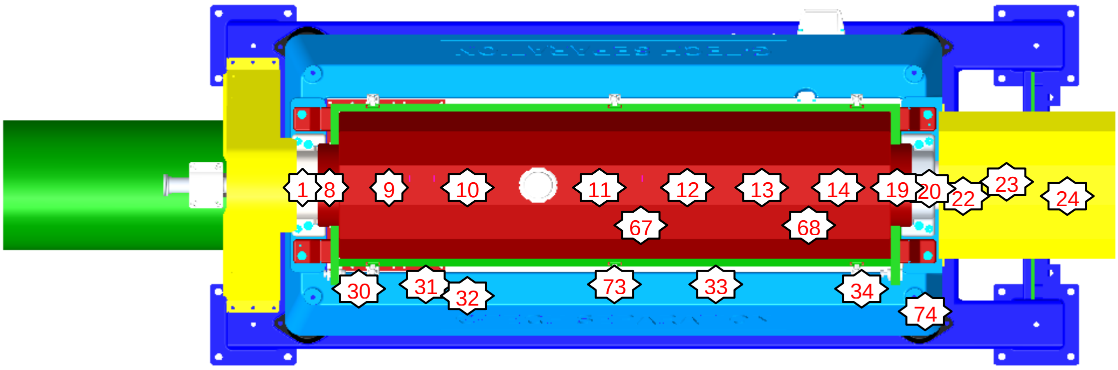


Figure 25 Measurement points on top of decanter

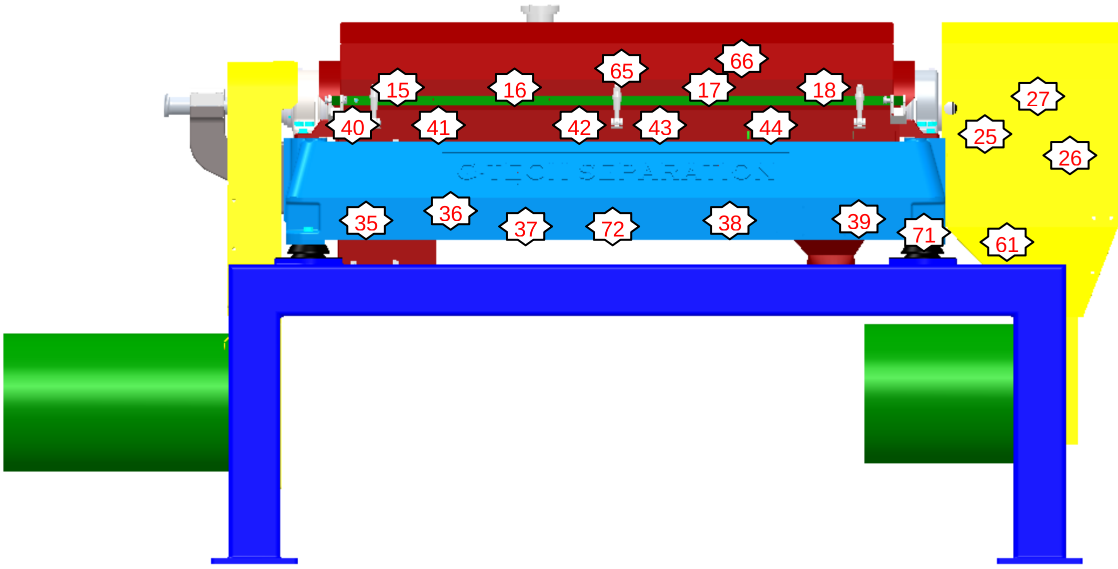


Figure 26 Measurement points on side of decanter

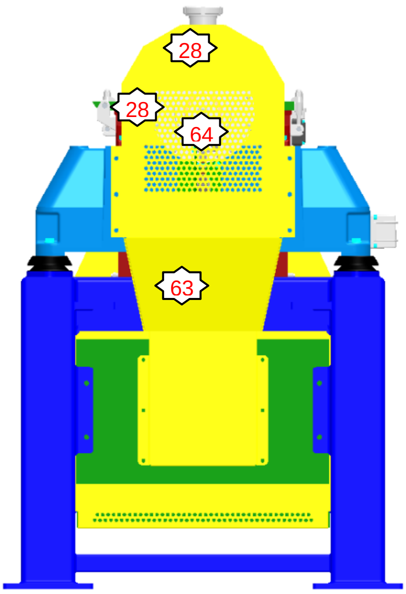


Figure 27 Measurement points on end of gearbox guard

2.10 Conclusion

The following parameters have been established for measuring the sound power of the decanter using sound intensity techniques:

- The main motor controller is set at 48.7 Hz.
- The back motor controller is set at 47.8 Hz.
- Exit ports are to be covered.
- The 12.5 mm microphone spacer is to be used for sound intensity measurements.
- The sound intensity windscreen is to be used.
- The room is to have the standard amount of sound absorption in the room, as illustrated in Figures 4 and 5.

The quality of the measurements was considered to be good for the 80 to 10,000 Hz one-third octave bands. The measurements for the 25 and 31.5 Hz one-third octave frequency bands are erratic and will not be presented in subsequent results. The 40 to 63 Hz one-third octave frequency bands will be presented but minor changes in measurement levels should not be used to infer that actual changes in the system have occurred.

Acceleration measurements are to be made at 1 Hz resolution over the frequency range of 1 to 1600 Hz, using a Hanning time weighting window, averaged of 177 measurements (45 seconds).

3 Decanter Evaluation

The sound power of a continuous discharge decanting centrifuge, hereafter referred to as a decanter, was determined using two sound intensity methods. The first method used was the sweep method and the second method was the point method. Both methods were used to determine which components of the decanter produce the higher noise levels. The sweep method was then used to determine the overall sound power of the decanter. The overall sound power will then be used to evaluate the effectiveness of various planned changes to the decanter carried out with the objective of reducing the sound power.

3.1 Decanter Sound Intensity Measurements

3.1.1 Contour Plots of Sound Intensity for the Decanter

Point measurements were made in order to measure the sound intensity over each face of the decanter. A virtual box of 3600 x 1400 x 1800 mm was created that enclosed the decanter, as shown in Figure 28. Each side of the box was divided into 200 x 200 mm squares. Fifteen second point sound intensity measurements were then taken at the centre of each of the 576 squares. The time required to complete the scan was over eight hours. From the results a contour plot of the sound intensity over the decanter surface was produced using MATLAB. The plots are shown in Table 2. The MATLAB program extrapolates measurements to estimate sound intensity levels along the edges. Extreme high and low intensity levels along the edges are not actual measurements and should be ignored. Figure 28 also shows the orientation of the decanter in the contour plots. The overall sound power is shown in Figure 29.

The 100 Hz one-third octave band contains most of the sound power, which includes the second harmonic of the driven frequency. The 50 to 1000 Hz one-third octave bands contain the sound powers over 76 dB and the 50 to 500 Hz one-third octave bands contain the sound powers over 80 dB. This indicates that the noise levels at the lower frequencies are the ones that need to be reduced to lower the sound power of the decanter. The sound power distribution is the same as for the component scan but the measurement levels are about 10 dB lower.

Vibrations of the decanter are generated by the rotating assembly which has a fundamental frequency of 54 Hz. The 50, 100, and 160 Hz one-third octave frequency bands contain the first three harmonics of the driven frequency. The one-third octave bands that do not contain a harmonic of the driven frequency (40, 63, 80, 125) are clearly lower than the surrounding bands that do contain a harmonic of the driven frequency. This suggests the rotating bowl was the primary source of the vibration of the decanter and the high noise levels are a consequence of the resonant vibration of structural parts.

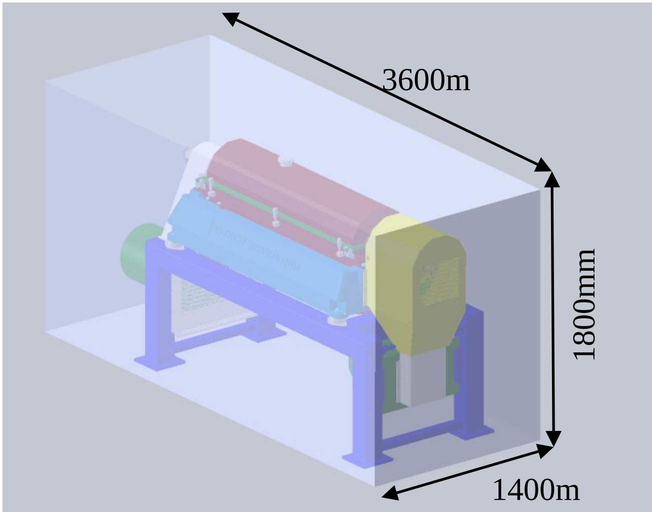


Figure 28 Virtual box over the decanter

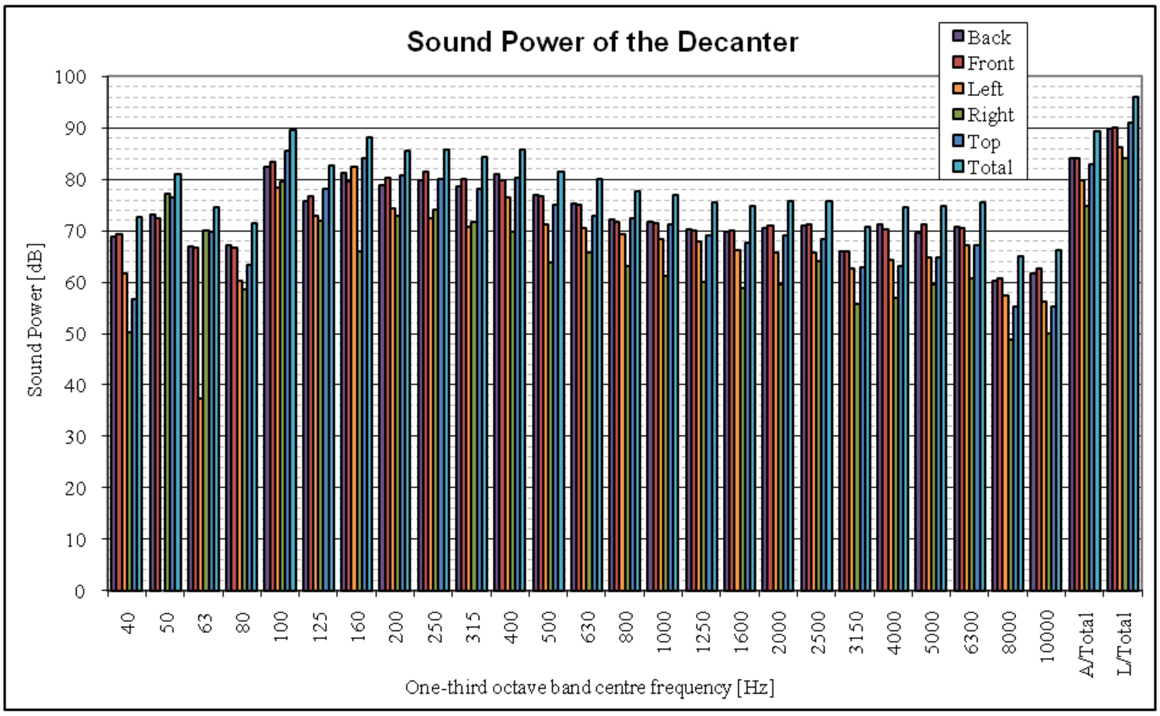
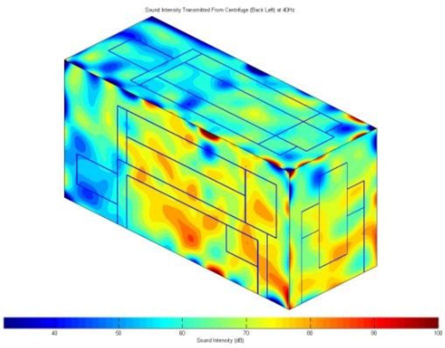
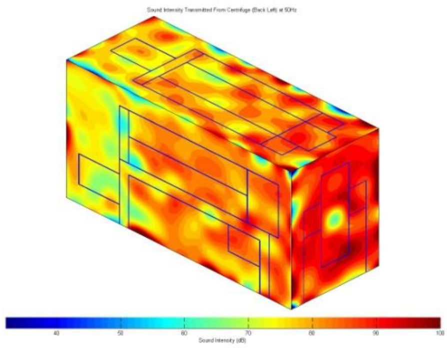
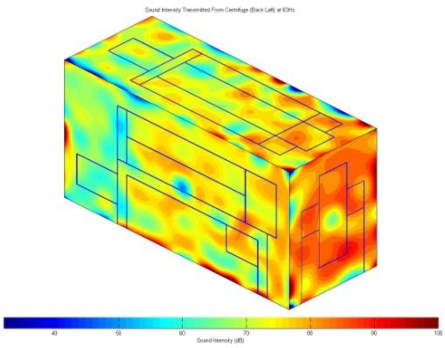
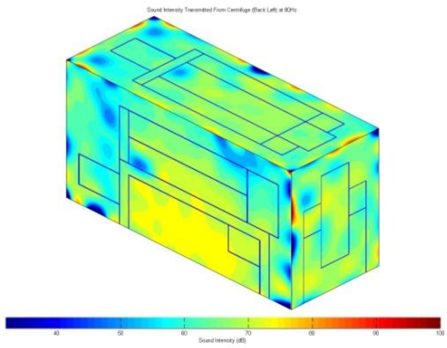
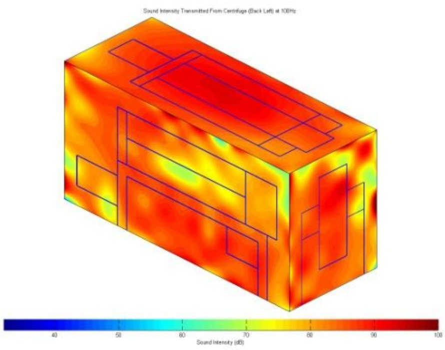
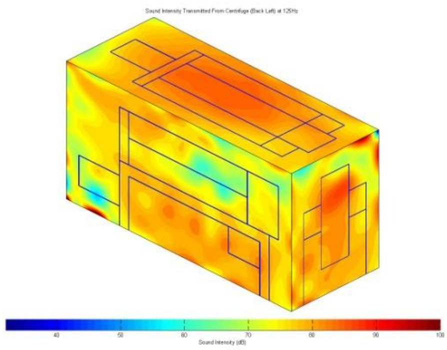


Figure 29 Sound power of decanter

Table 2 Sound intensity contours for the decanter

<table>
<tr><td colspan="2">Contour gradient scale used in all plots.

30 40 50 60 70 80 90 100 [dB]</td></tr>
<tr><td>
40 Hz one-third octave band</td><td>
50 Hz one-third octave band</td></tr>
<tr><td>
63 Hz one-third octave band</td><td>
80 Hz one-third octave band</td></tr>
<tr><td>
100 Hz one-third octave band</td><td>
125 Hz one-third octave band</td></tr>
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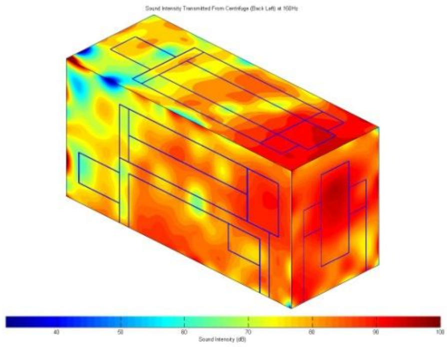
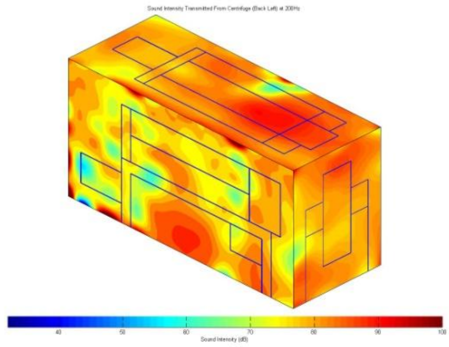
<td>

160 Hz one-third octave band</td>
<td>

200 Hz one-third octave band</td>
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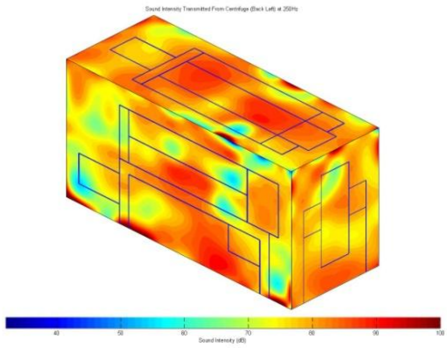
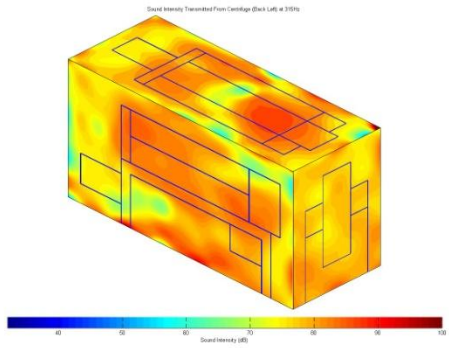
<td>

250 Hz one-third octave band</td>
<td>

315 Hz one-third octave band</td>
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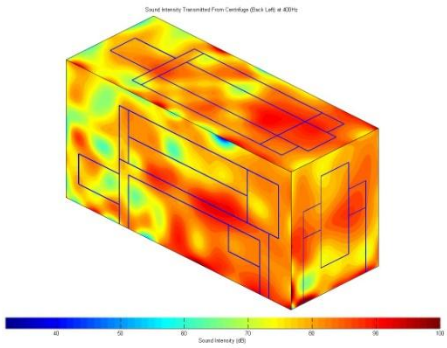
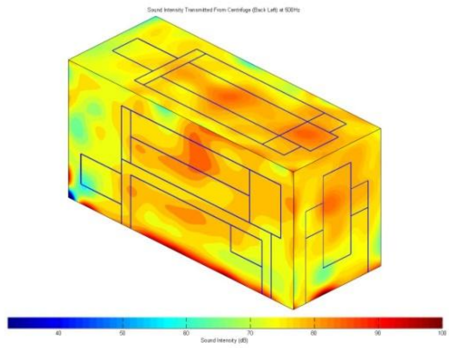
<td>

400 Hz one-third octave band</td>
<td>

500 Hz one-third octave band</td>
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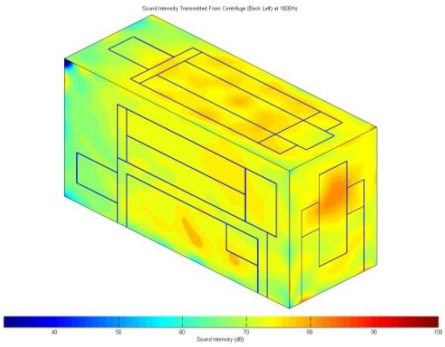
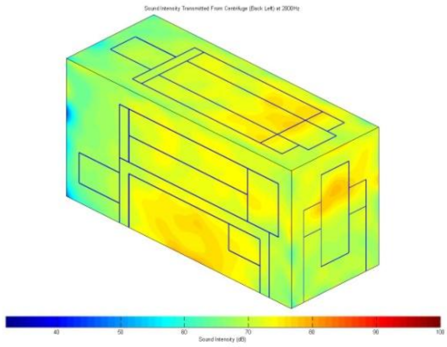
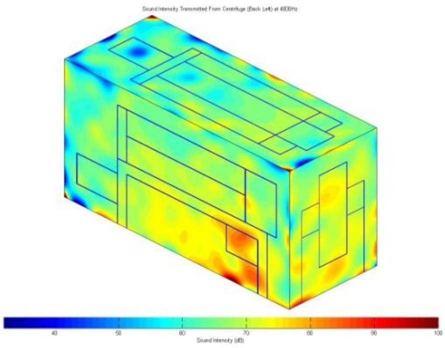
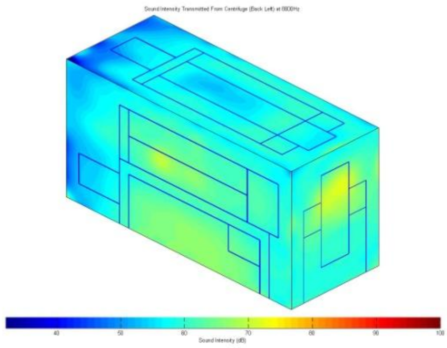
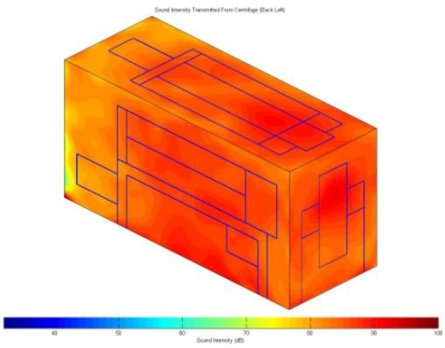
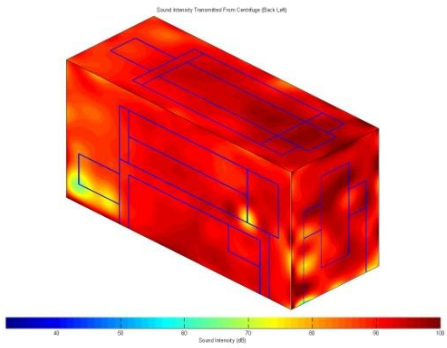
 1000 Hz one-third octave band	 2000 Hz one-third octave band
 4000 Hz one-third octave band	 8000 Hz one-third octave band
 A-weighted sound power total	 Sound power total

3.1.2 Evaluation of Decanter Components by Sweep Method

The purpose of this test was to identify which components of the decanter were the major contributors to the sound power of the decanter. The sound power of the decanter was determined by the sweep method and was carried out according to ISO 9614-2. The sweep method involves enclosing the machine in a number of scan surfaces. The intensity probe is then swept over each surface to determine the average sound intensity of each surface. This method allowed the identification of the contribution of the various components to the overall sound power by scanning over the surfaces associated to each component. Figures 30 to 32 show the decanter and how the various components were labelled and scanned. The scanned surfaces did not form a simple box shape but were more like a pyramid. The front and back surfaces step in above the base and at the main motor. The top surface steps down over the base and main motor. The right side steps in above the main motor. The time required to complete the scan was about two hours. Due to the large number of surfaces scanned the analysis was also time consuming.

The resulting sound power measurements are shown in Figure 33. The results show that most of the sound power is coming from the frame and opening. This is to be expected as this represents the largest area of the decanter. The next three components with the highest sound power are the base, hopper and gearbox guard. The main motor and main motor guard are not significant contributors to the overall sound power of the decanter.

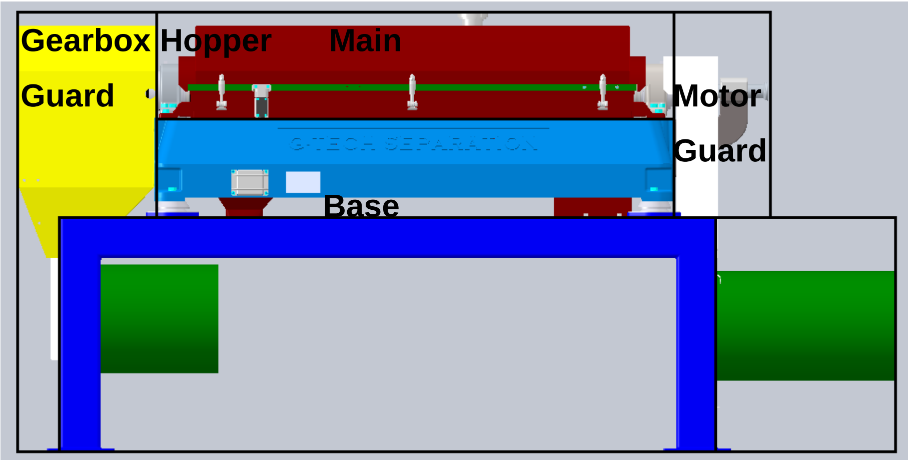


Figure 30 Front view of the decanter showing component positions

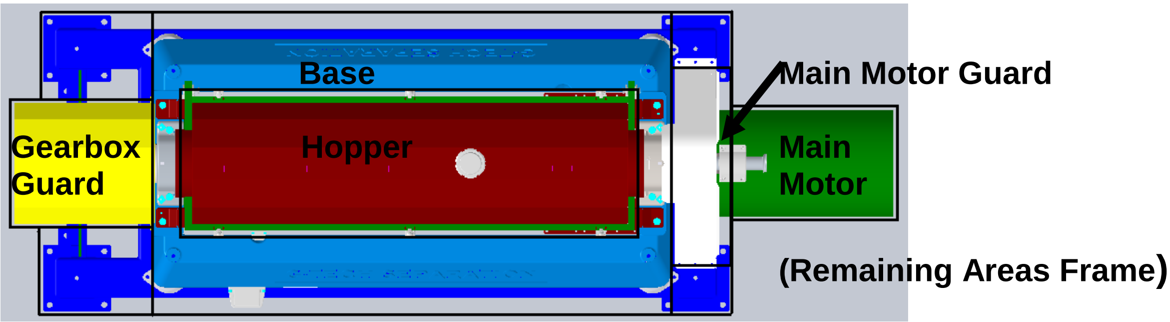


Figure 31 Top view - component positions

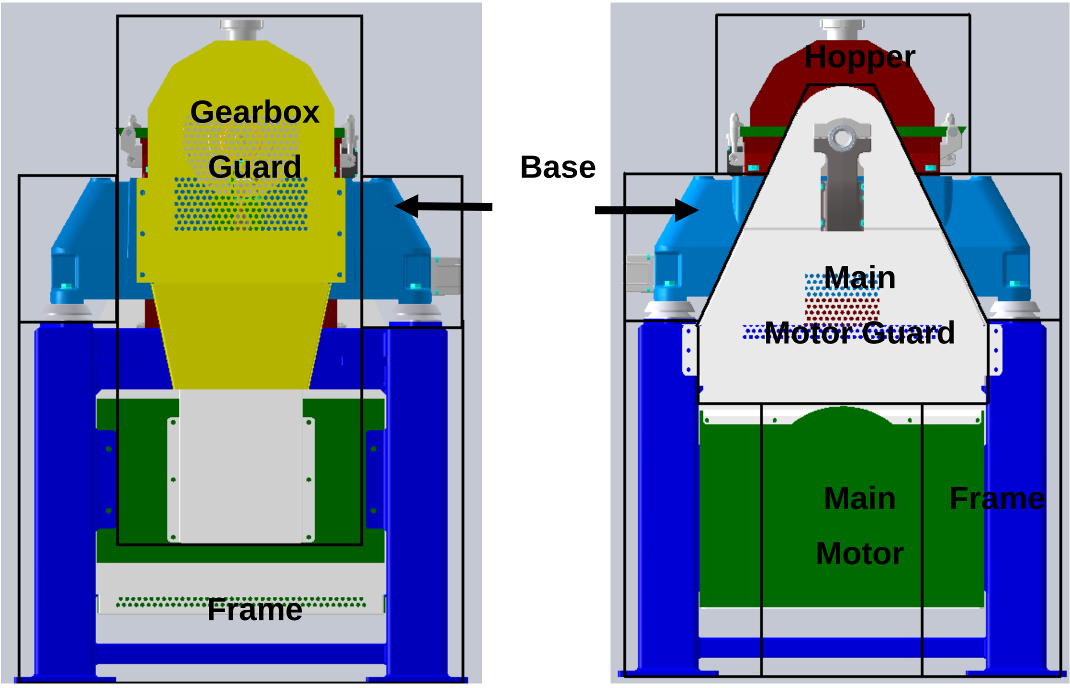


Figure 32 Left and right view - component positions

The 100 Hz one-third octave band contains most of the sound power. The bowl rotates at 54 Hz so the 100 Hz one-third octave band contains the second harmonic of the driven frequency. The 50 to 1250 Hz one-third octave bands contain the total sound powers over 86 dB and the 50 to 500 Hz one-third octave bands contain the total sound powers over 90 dB. This indicates that the noise levels at the lower frequencies are the ones that need to be reduced in order to lower the sound power of the decanter.

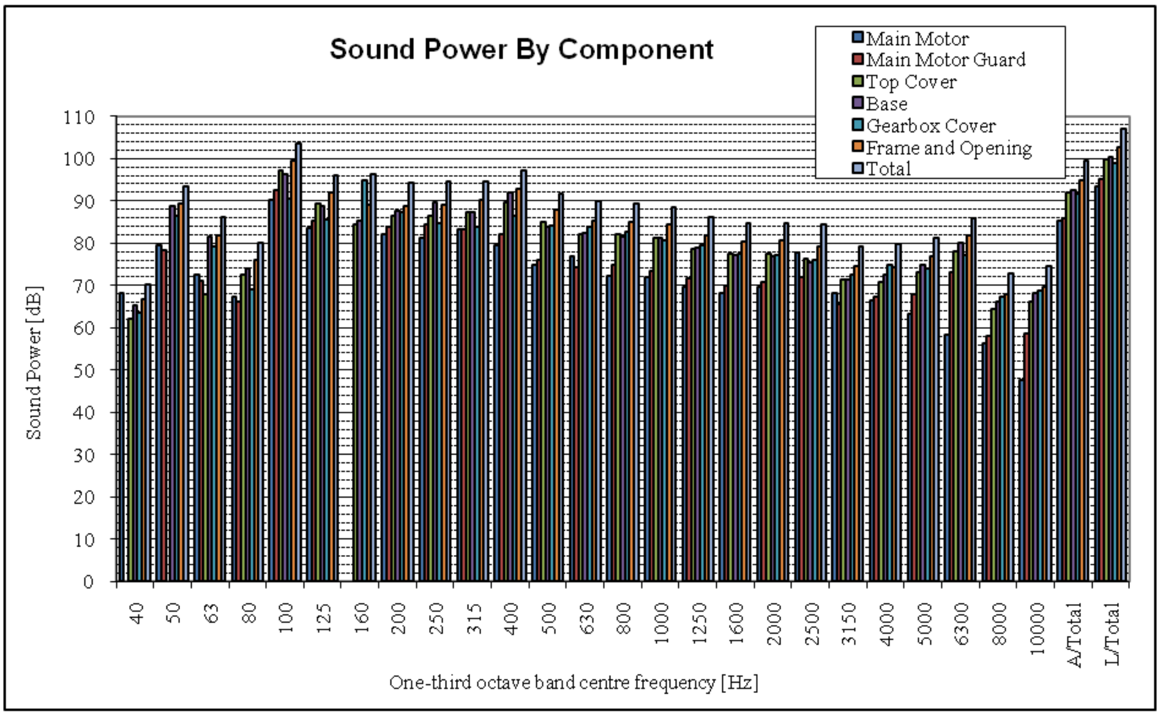


Figure 33 Sound power of components

3.1.3 Decanter Sound Power Assessment

From assessing various techniques for measuring the sound intensity it was determined that the sweep method was most effective as it produced accurate, repeatable and quick results. A virtual box was created enclosing the decanter and the surfaces of this box were divided into sub regions as detailed in Section 2.7. The decanter was scanned on two separate days and the results are shown in Tables 3 to 5 and Figure 34. Only 40 minutes was required to scan the entre decanter, which was significantly quicker than the methods described in Sections 3.1.1 and 3.1.2.

The 100 and 160 Hz one-third octave bands contain most of the sound power. These two bands contain the second and third harmonic of the driven frequency. The 50 to 630 Hz one-third octave bands contain the sound powers over 86 dB and the 50 to 400 Hz one-third octave bands contain the sound powers over 90 dB. These results show the same pattern as the component and point scan.

Table 3 Overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	92.7	91.4	91.9	87.9		89.4	91.5	93.0	87.1	91.1
OC2	93.1	91.6	93.0	88.2		89.4	91.7	93.1	87.4	91.5
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	86.7	92.5	93.0	96.0		75.9	91.0	91.2	91.0	91.4
OC2	87.4	92.5	92.9	95.1		80.7	91.7	91.4	90.3	91.4
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	93.4	97.8	96.2	91.9	94.6			89.9	90.5	90.2
OC2	93.8	97.0	95.5	92.0	94.2			88.9	89.6	89.3
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	94.5	91.3	93.9	90.2	92.6					91.9
OC2	93.7	92.8	92.1	88.7	92.0					91.8

Table 4 Sound intensity measurements for the 100 Hz one-third octave band [dB]

One-third octave band centre frequency:				100	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	81.0	83.5	88.4	82.2		76.9	77.0	87.7	79.0	84.3
OC2	81.0	83.2	89.5	83.2		0.0	78.6	88.6	81.7	85.2
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	83.9	85.2	79.3	81.2		76.6	85.3	79.5	80.2	82.5
OC2	84.9	85.7	79.0	80.6		78.9	84.9	80.9	79.7	82.9
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	79.0	89.3	86.6	85.6	86.4			87.4	86.2	86.8
OC2	83.1	87.9	85.4	85.2	85.8			86.3	84.8	85.6
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	85.5	86.4	88.3	88.8	87.6					85.3
OC2	84.0	86.8	88.8	84.7	86.8					85.1

Table 5 Sound intensity measurements for the 160 Hz one-third octave band [dB]

One-third octave band centre frequency:				160	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	89.0	80.0	78.0	83.2		82.5	74.8	84.7	82.6	82.9
OC2	89.5	81.5	75.8	82.3		83.7	72.6	83.0	79.1	82.4
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	0.0	79.2	87.0	94.4		0.0	0.0	81.2	85.6	84.6
OC2	0.0	76.5	85.8	93.2		0.0	76.1	75.3	84.6	83.6
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	91.9	96.1	92.2	86.7	91.8			0.0	71.2	0.0
OC2	91.6	94.9	90.8	87.1	91.0			0.0	76.8	66.1
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	90.1	71.0	88.0	0.0	85.3					85.5
OC2	90.3	82.2	71.6	0.0	83.4					84.6

The two scans show very consistent results for each area scanned and for the overall totals of the sides of the decanter. This indicates that the results are repeatable and suitable for assessing sound power changes caused by subsequent modifications. The total sound power of the decanter is 104 dB.

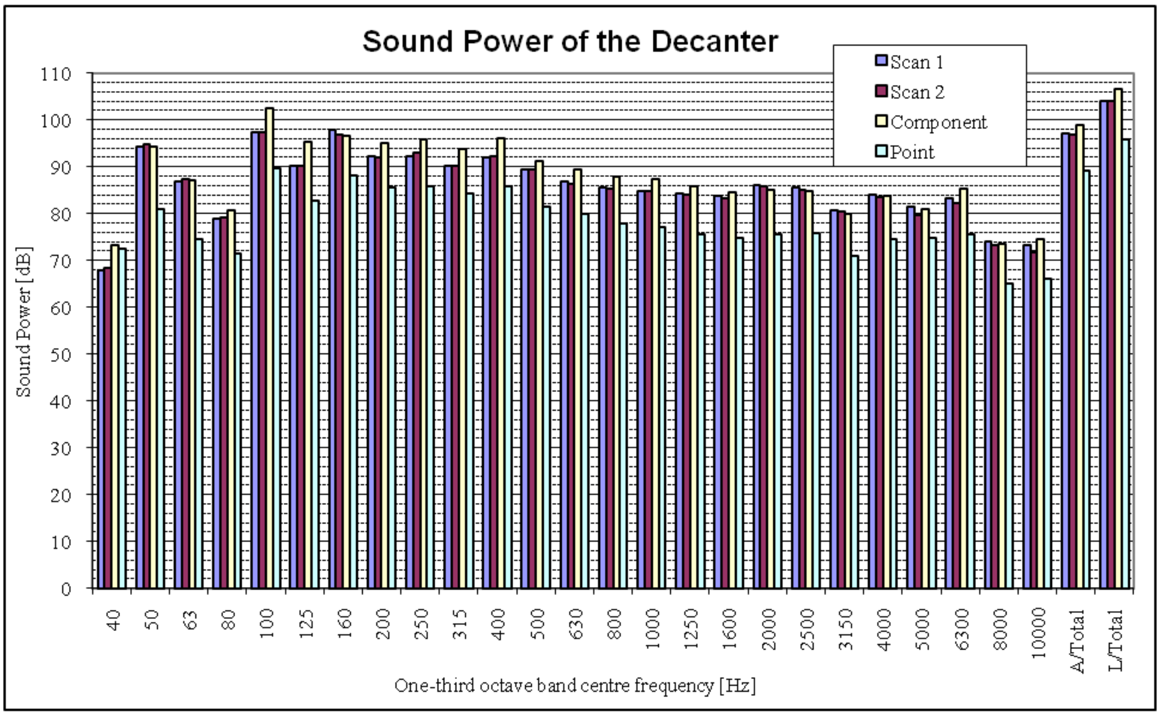


Figure 34 Sound power of all four tests

3.2 Comparison of Sound Intensity Measurements

The sound powers measured for all four scans are shown in Figure 34. The component scan gave the highest sound power measurements and the point scan gave the lowest. The reason that the component scan had the highest measurements was probably because the sound pressures measured near corners were higher due to reflections (see Figure 35 (a)), which result in higher intensity measurements. Alternatively the sound power was higher for the component scan because this scan was done before the other scans, while the decanter had not been fully 'run-in'. A significant amount of running was completed before the remaining three scans were done.

The reason that the point scan of the decanter resulted in lower sound power measurements was probably due to the measurement point locations being predominantly aligned with the corners of decanter surfaces as compared to the middle of surfaces, see Figure 35 (b). When a plate vibrates the area of the plate can be divided into two types of regions: nodal areas where the plate does not vibrate and remaining areas were vibrating occurs that produce sound. When a plate joins another plate at a corner, the corner will be constrained and become a nodal area for all frequencies. The alignment of the point measurements with predominantly corners, compared to mid-surfaces, accounts for lower sound power measurements across all frequencies.

a Measuring Near Corners | **b** Corner scan verses Mid-surface scan

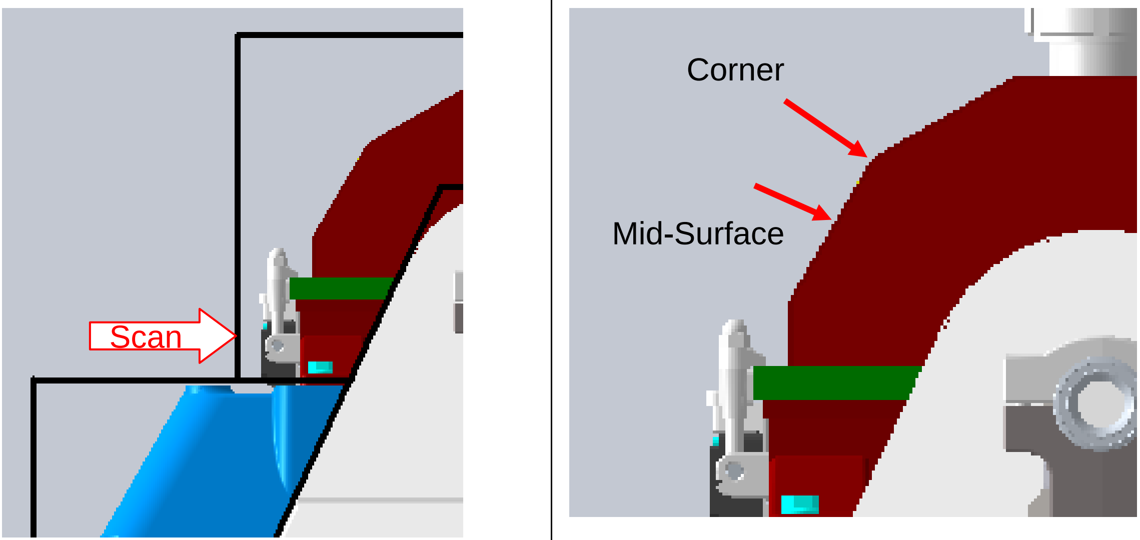


Figure 35 (a-b) Component scan

3.3 Acceleration Measurements

The acceleration measurements were taken in accordance with Section 2.9. The overall acceleration is the combination of all the vertical, lateral and additional measurement points. The results of the acceleration measurements are shown in Figures 36 to 38. The labels along the frequency axis have been done to match the harmonic frequencies of the rotating bowl. The graphs show that the vibrations within the base, gearbox guard and hopper are highest at the harmonic frequencies of the rotating bowl.

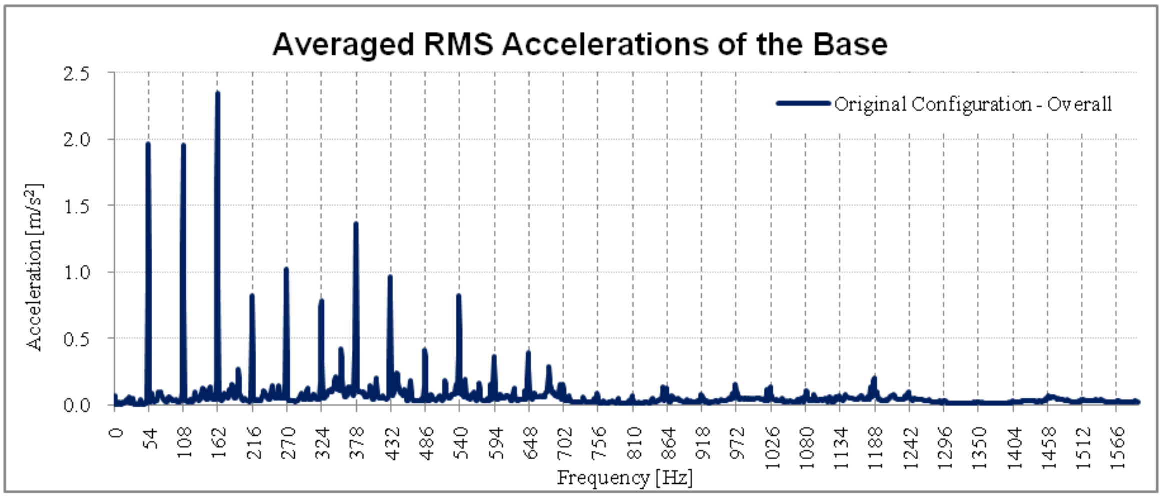


Figure 36 Accelerations of the base – overall

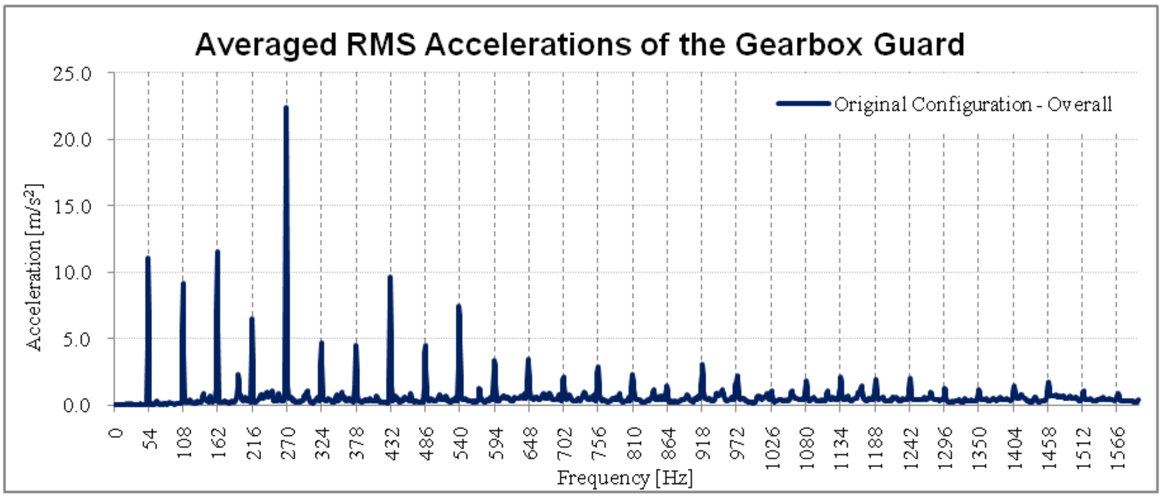


Figure 37 Accelerations of the gearbox guard – overall

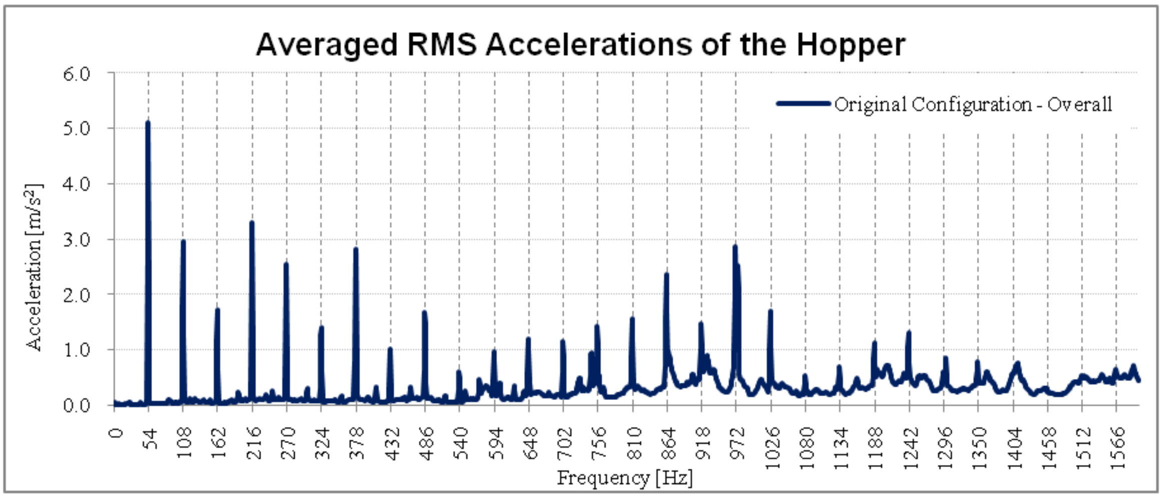


Figure 38 Accelerations of the hopper – overall

3.4 Conclusion on Measurements of the Original Decanter Configuration

The sweep method used to produce Scans 1 and 2 provides an excellent base against which subsequent measurements can be compared. The repeatability of this method ensures that minor changes in sound power can be measured. The short scan time allows time to do repeat measurements and confirm changes in the sound power. The total sound power of the decanter was found to be 104 dB and the one-third frequency bands of main interest are 50 to 630 Hz. The acceleration measurements show that the vibrations within the base, gearbox guard and hopper are highest at the harmonic frequencies of the rotating bowl.

4 Modification – Isolation of Gearbox Guard

The first step undertaken to reduce the sound power generated from the decanter was to isolate the gearbox guard from the base. The base of the decanter serves to support most of the major components of the decanter. The main sources of vibration are from the rotating assembly which is mounted to the top of the base at each end (see Figure 39). The gearbox guard is mounted to the end of the base near one of the main bearings for the bowl (see Figure 40 (a)). The vibrations generated in the bowl have a direct path to the gearbox guard. The gearbox guard was unbolted from the base and mounted to a bracket that is bolted to the sub frame (see Figure 40 (a)). The base sits on isolation feet that are mounted to the sub frame. The vibrations generated from the rotating assembly would now need to pass through the isolation feet to get to the gearbox guard. This new configuration will be referred to as OC-GI.

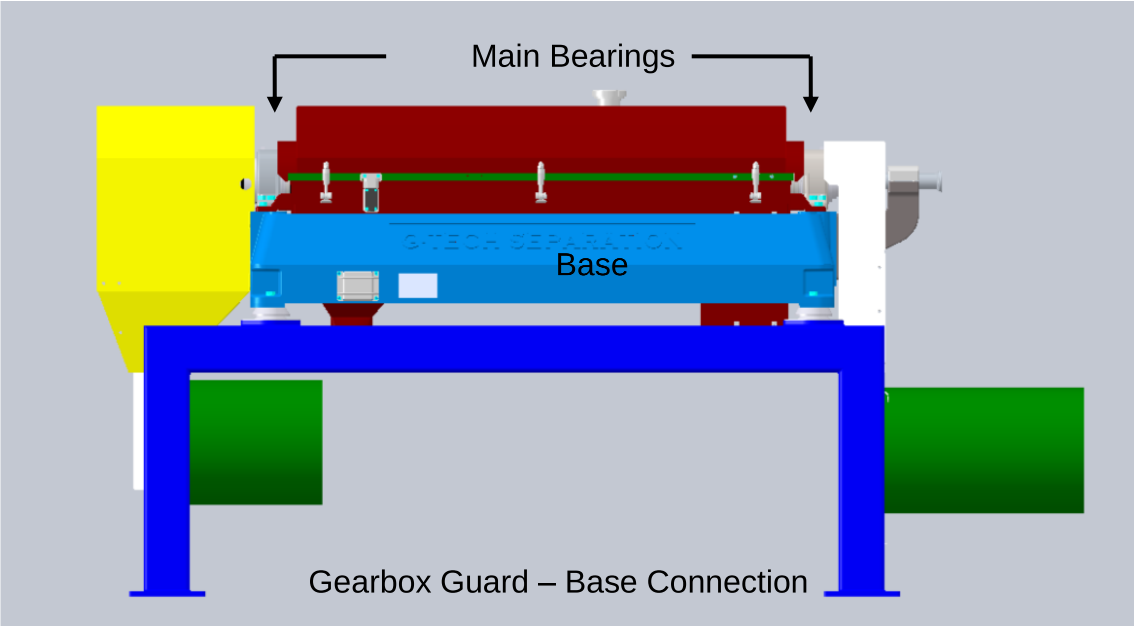


Figure 39 Decanter - standard configuration

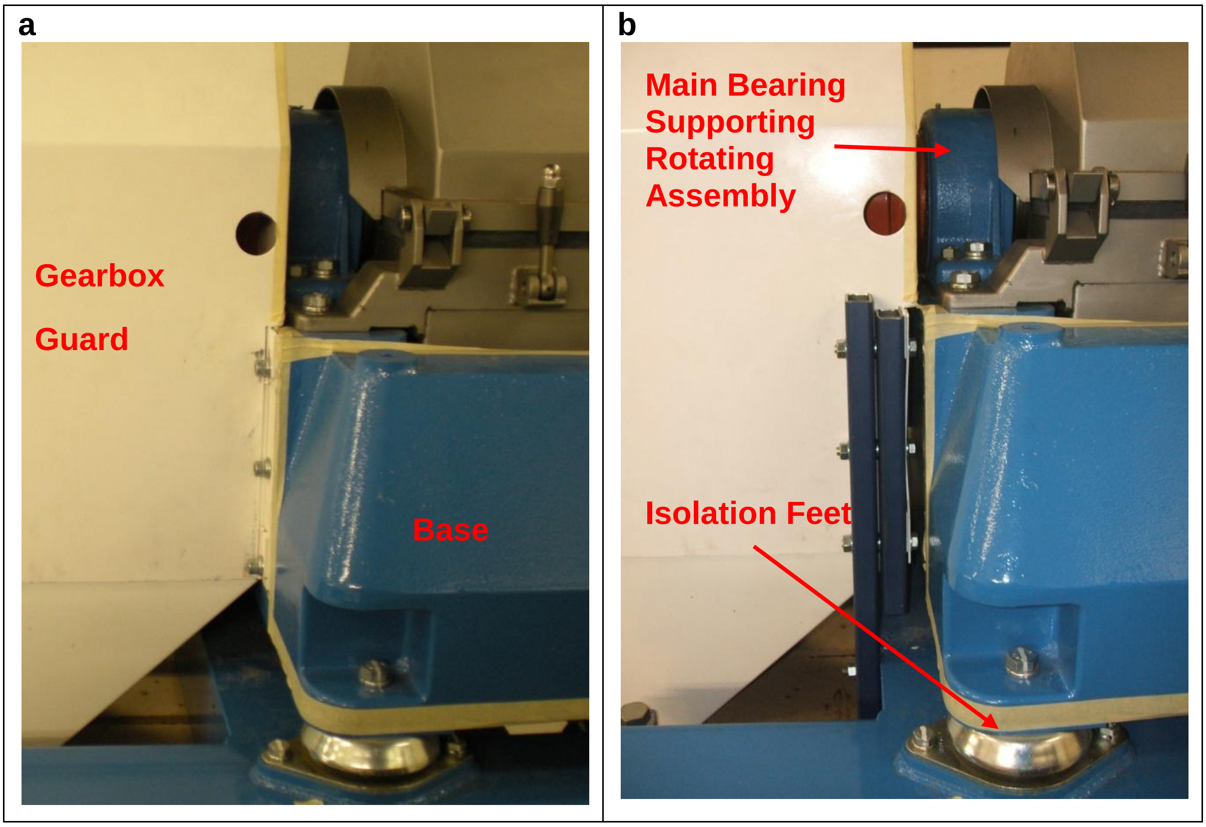


Figure 40 (a-b) Gearbox guard mounting

4.1 Results

The results of the two sound intensity scans are shown in Figure 41. The scans were undertaken by the method described in Section 2.7. The sound power of the decanter was 104.7 dB and the A-weighted sound power level was 99.1 dB(A). Vibration measurements were taken in accordance with Section 2.9. The overall results for the vibration acceleration are shown in Table 6, with the spectrum shown in Figure 42. The overall vibration levels have significantly reduced primarily due to a significant reduction in the vibration levels in the gearbox guard.

Table 6 Overall acceleration results [m/s2]

Configuration	Base	Hopper	GB Guard	Total
Original	0.15	0.46	1.26	0.70
OC-GI	0.17	0.45	0.12	0.33

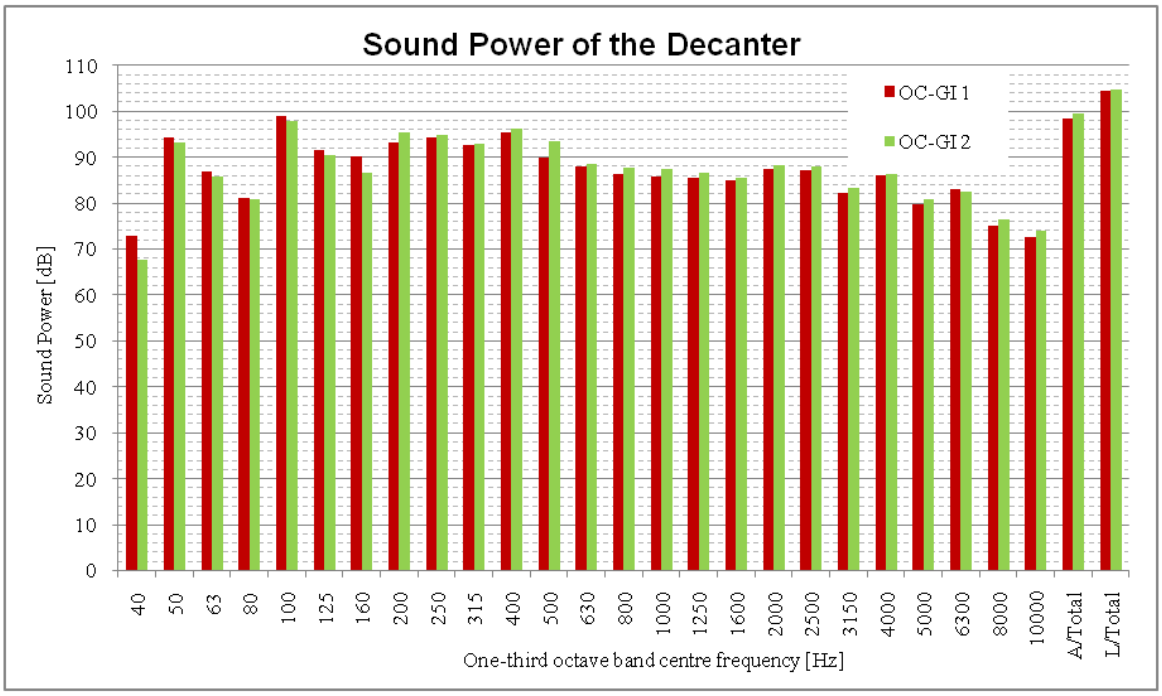


Figure 41 Sound power of the decanter

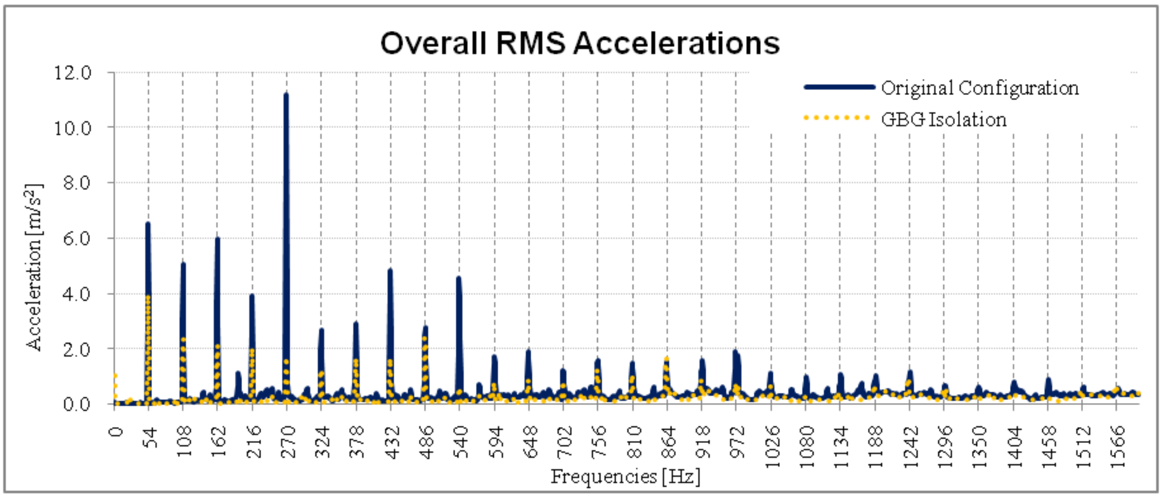


Figure 42 Accelerations of the base - overall

4.2 Comparison of Sound Intensity Measurements

A virtual box was created enclosing the decanter and the surfaces of this box were divided into sub regions as detailed in Section 2.7. A comparison of the sound powers of the decanter with the new base and with the original configuration is shown in Figure 43 and Tables 7 and 8. The overall sound power of the decanter increased 0.7 dB to 104.7 dB. The A-weighted sound power increased 1.1 dB to 99.1 dB(A). The main differences in sound power is the 8.7 dB decrease in the 160 Hz one-third octave band and the increases in sound power in all the one-third octave bands above 160 Hz.

From Table 7, it is evident that the sound intensity from the regions directly associated to the gearbox guard have been significantly reduced (Front 1: -2.5 dB, Back 4: -5.1 dB, Left 2: -8.4 dB, Top 1: -3.8 dB). It is also evident that nearly all the other regions have shown an increase in sound intensity and the overall sound power has increased 0.7 dB. The reason that the sound intensity of the decanter did not decrease is probably due to the energy that was used to vibrate the gearbox guard now being used to increase the vibrations in the base, hopper. The results show that isolating the gearbox guard alone does not decrease the sound power of the decanter.

The 160 Hz one-third octave band showed the largest decrease in sound power due to isolating the gearbox guard. Table 8 shows the sound intensity distribution for the 160 Hz one-third octave band. The 8.7 dB decrease corresponds to the only one-third octave band that has the gearbox guard as the dominant source, see Figure 33. The gearbox guard produced most of its sound power in the 160 Hz one-third octave band before being modified. These facts explain the significant reduction in sound power in the 160 Hz one-third octave band.

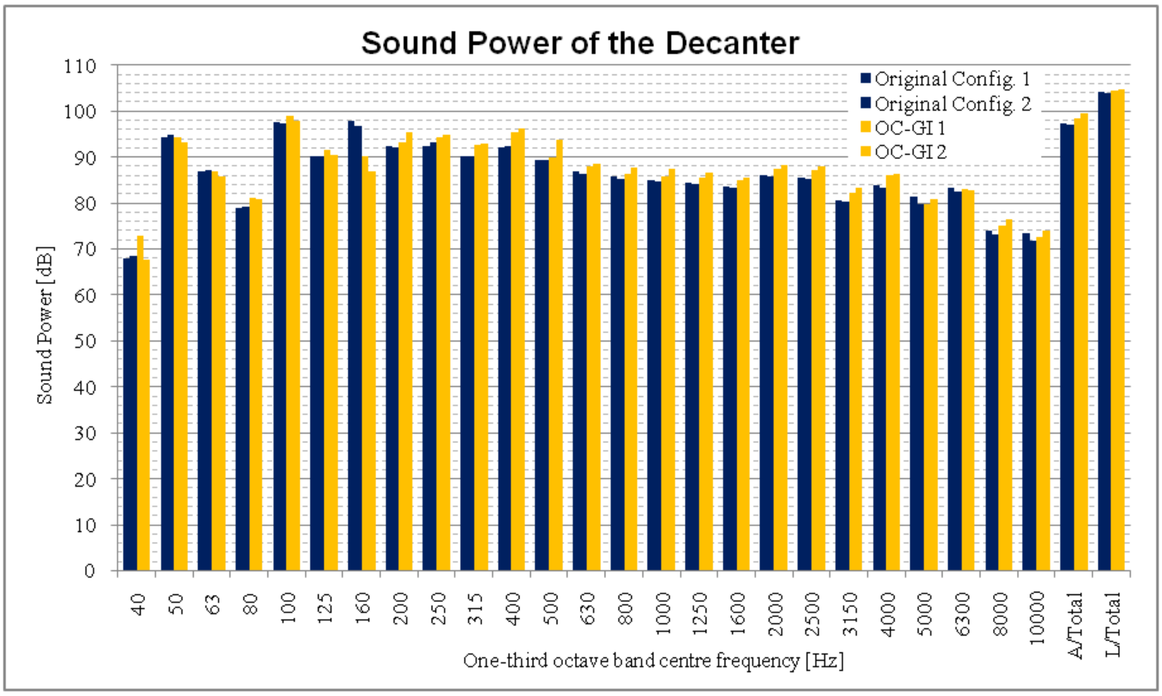


Figure 43 Sound power of decanter

Table 7 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	92.7	91.4	91.9	87.9		89.4	91.5	93.0	87.1	91.1
OC2	93.1	91.6	93.0	88.2		89.4	91.7	93.1	87.4	91.5
OC-GI1	89.8	93.2	94.5	88.2		90.2	93.2	94.6	88.1	92.5
OC-GI2	91.1	92.7	94.3	87.7		91.6	93.7	93.7	85.2	92.3
Ave Diff.	⇩ -2.5	⇧ 1.5	⇧ 2.0	⇨ -0.1		⇧ 1.5	⇧ 1.9	⇧ 1.1	⇨ -0.6	⇧ 1.1
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	86.7	92.5	93.0	96.0		75.9	91.0	91.2	91.0	91.4
OC2	87.4	92.5	92.9	95.1		80.7	91.7	91.4	90.3	91.4
OC-GI1	91.3	94.4	94.1	90.1		89.6	94.3	92.3	88.4	92.7
OC-GI2	91.3	95.1	94.8	90.8		88.6	94.4	93.8	89.5	93.2
Ave Diff.	⇧ 4.2	⇧ 2.3	⇧ 1.5	⇩ -5.1		⇧ 10.8	⇧ 3.0	⇧ 1.7	⇩ -1.7	⇧ 1.5
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	93.4	97.8	96.2	91.9	94.6			89.9	90.5	90.2
OC2	93.8	97.0	95.5	92.0	94.2			88.9	89.6	89.3
OC-GI1	88.7	89.1	88.4	90.4	89.7			91.3	91.4	91.4
OC-GI2	87.8	89.0	89.8	91.0	90.2			91.0	90.8	90.9
Ave Diff.	⇩ -5.3	⇩ -8.4	⇩ -6.8	⇩ -1.3	⇩ -4.5			⇧ 1.8	⇧ 1.1	⇧ 1.4
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	94.5	91.3	93.9	90.2	92.6					91.9
OC2	93.7	92.8	92.1	88.7	92.0					91.8
OC-GI1	91.2	94.8	95.4	91.5	94.0					92.6
OC-GI2	89.4	94.3	95.1	91.0	93.4					92.6
Ave Diff.	⇩ -3.8	⇧ 2.5	⇧ 2.3	⇧ 1.8	⇧ 1.4					⇨ 0.7

Table 8 Comparison of sound intensity measurements for the 160 Hz one-third octave band [dB]

One-third octave band centre frequency:				160	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	89.0	80.0	78.0	83.2		82.5	74.8	84.7	82.6	82.9
OC2	89.5	81.5	75.8	82.3		83.7	72.6	83.0	79.1	82.4
OC-GI1	80.8	81.3	80.9	76.8		75.4	78.7	82.0	0.0	79.4
OC-GI2	75.3	77.7	77.6	66.4		72.9	77.3	75.6	0.0	75.4
Ave Diff.	⇩ -11.2	⇩ -1.3	⇧ 2.3	⇩ -11.2		⇩ -8.9	⇧ 4.3	⇩ -5.1	-	⇩ -5.2
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	0.0	79.2	87.0	94.4		0.0	0.0	81.2	85.6	84.6
OC2	0.0	76.5	85.8	93.2		0.0	76.1	75.3	84.6	83.6
OC-GI1	0.0	83.3	78.4	0.0		0.0	81.7	77.7	67.4	78.4
OC-GI2	69.7	76.4	74.1	68.1		72.0	76.4	74.9	69.7	74.1
Ave Diff.	-	⇧ 2.0	⇩ -10.2	⇩ -25.7		-	⇧ 3.0	⇩ -1.9	⇩ -16.6	⇩ -7.9
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	91.9	96.1	92.2	86.7	91.8			0.0	71.2	0.0
OC2	91.6	94.9	90.8	87.1	91.0			0.0	76.8	66.1
OC-GI1	76.4	73.5	62.5	78.4	76.8			63.6	81.4	78.8
OC-GI2	71.5	73.9	71.8	77.0	75.5			63.4	75.0	72.6
Ave Diff.	⇩ -17.8	⇩ -21.8	⇩ -24.4	⇩ -9.2	⇩ -15.2			-	⇧ 4.2	⇧ 9.5
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	90.1	71.0	88.0	0.0	85.3					85.5
OC2	90.3	82.2	71.6	0.0	83.4					84.6
OC-GI1	73.7	71.4	77.7	73.0	74.8					78.2
OC-GI2	74.3	73.6	75.1	71.2	73.8					74.5
Ave Diff.	⇩ -16.2	⇩ -4.1	⇩ -3.4	-	⇩ -10.0					⇩ -8.7

4.3 Comparison of Vibration Acceleration Measurements

Figures 44 to 52 show the vibration acceleration measurement results of the three main components of the decanter under investigation.

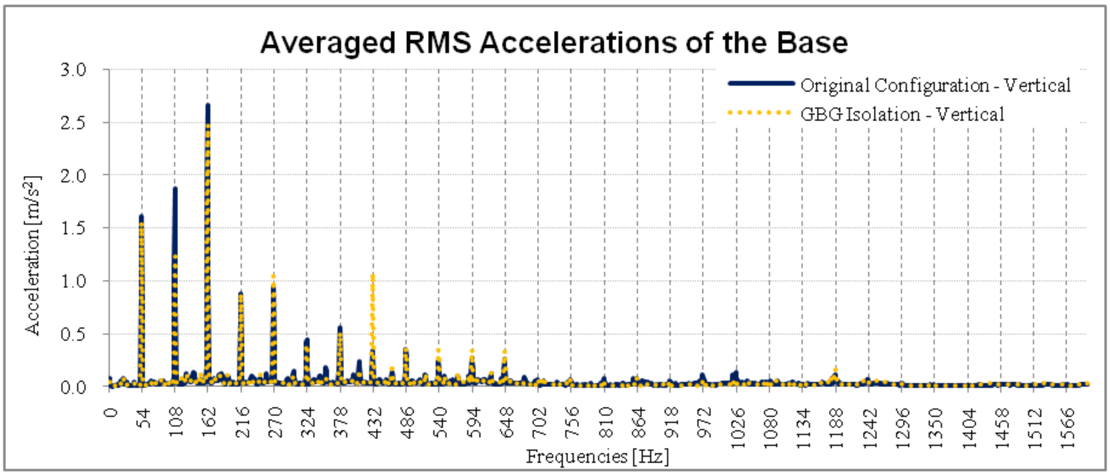


Figure 44 Accelerations of the base – vertical

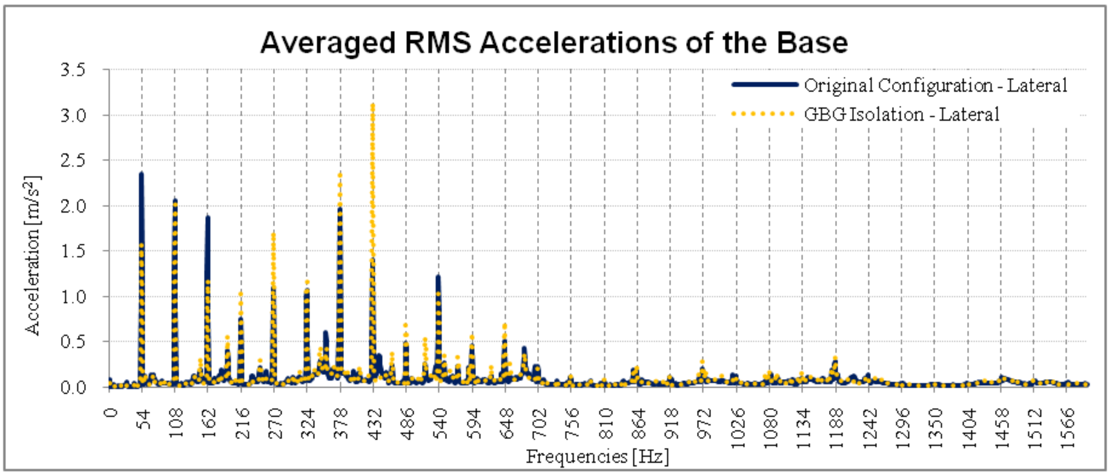


Figure 45 Accelerations of the base – lateral

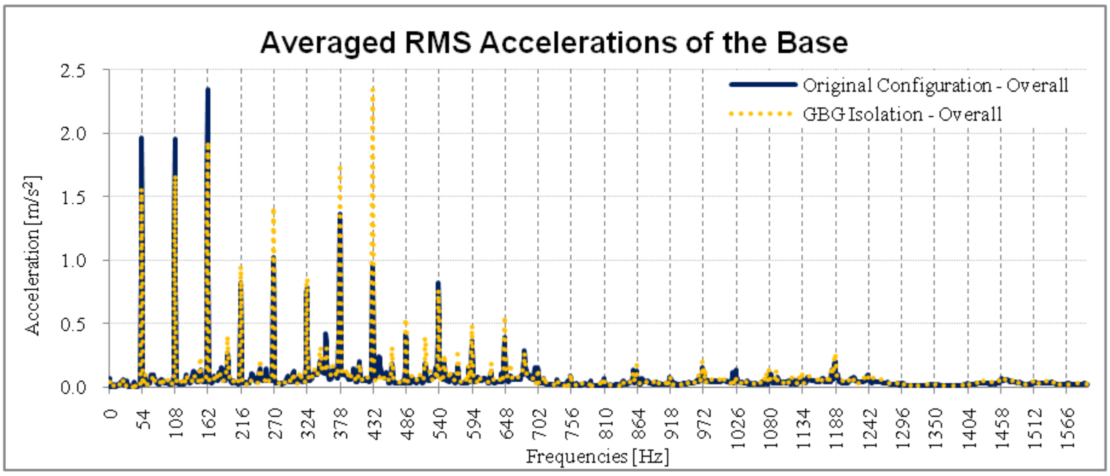


Figure 46 Accelerations of the base - overall

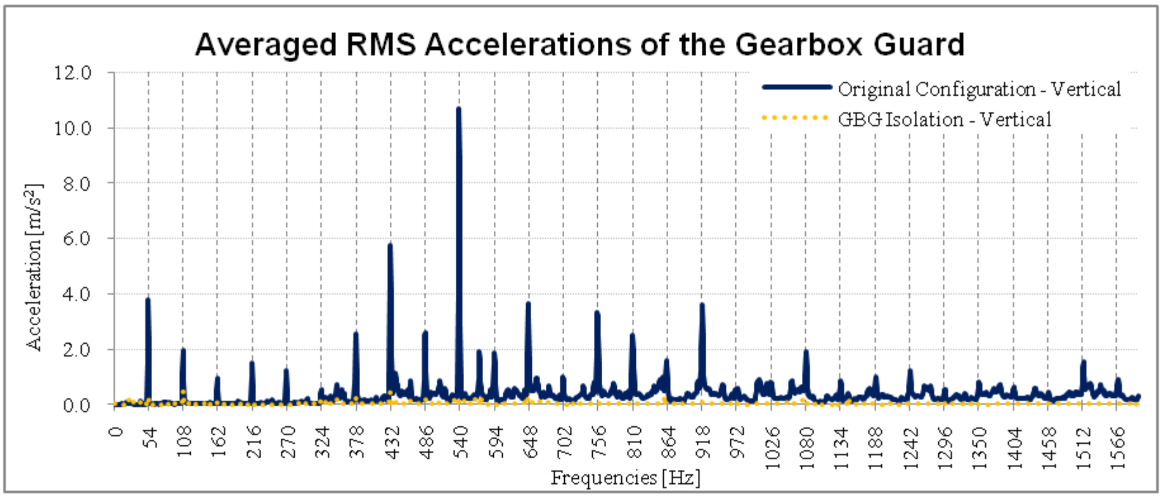


Figure 47 Accelerations of the gearbox guard – vertical

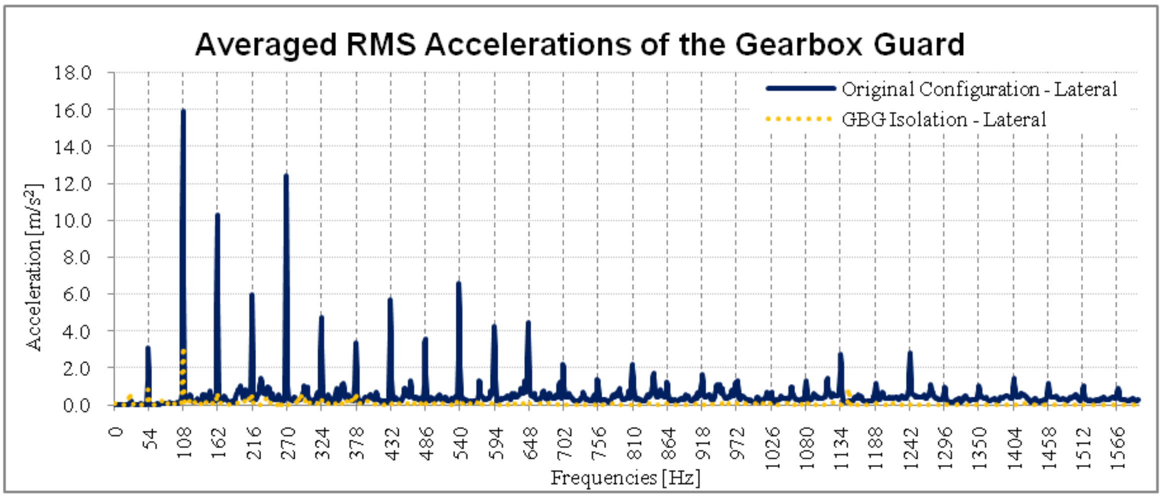


Figure 48 Accelerations of the gearbox guard – lateral

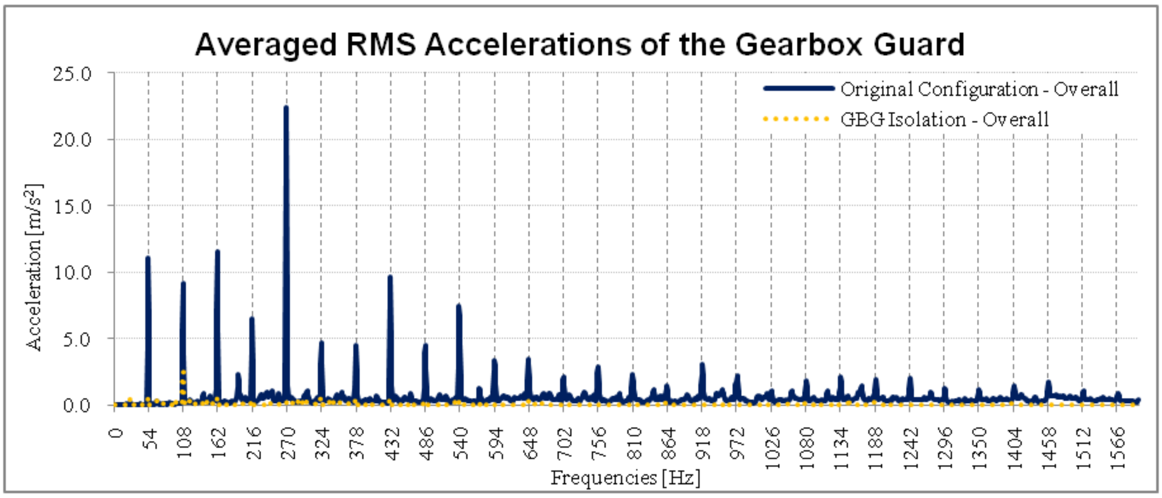


Figure 49 Accelerations of the gearbox guard – overall

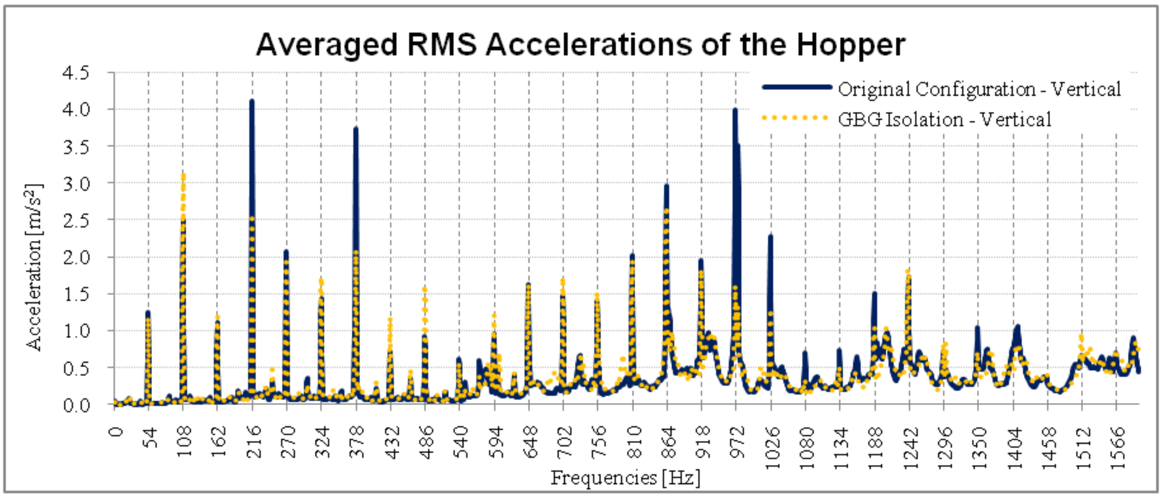


Figure 50 Accelerations of the hopper – vertical

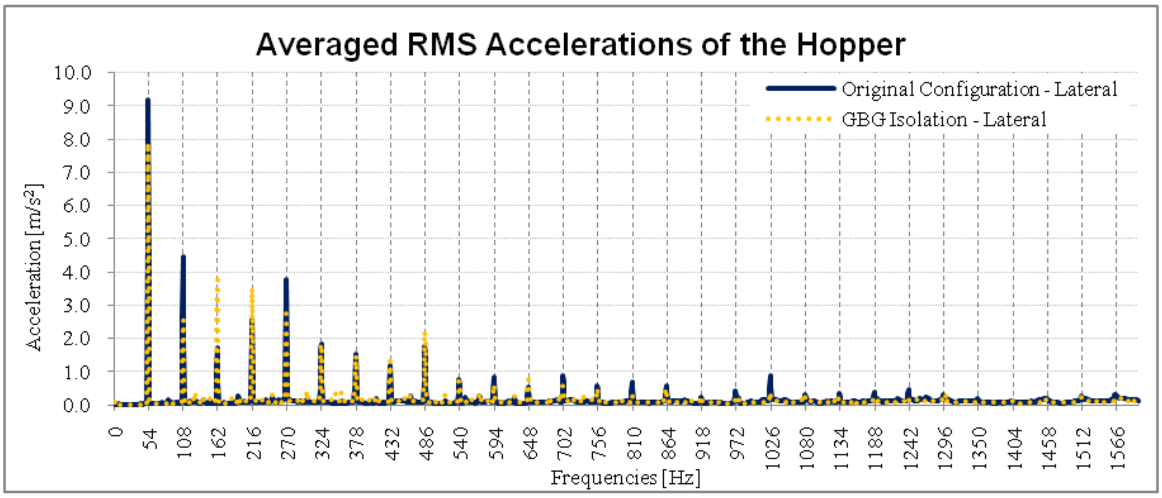


Figure 51 Accelerations of the hopper – lateral

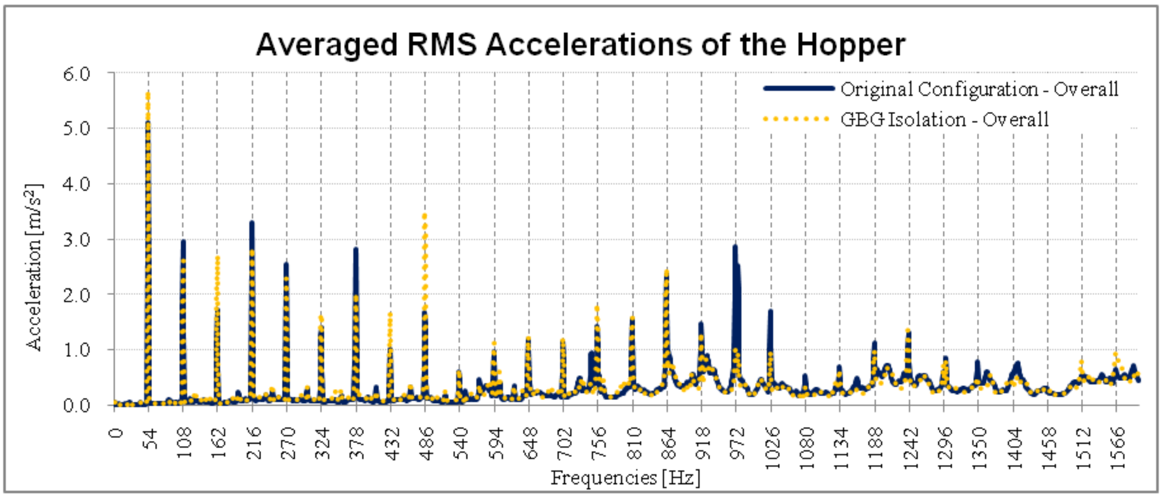


Figure 52 Accelerations of the hopper – overall

The total acceleration at each point was calculated by taking the 'root of the mean of the squares' (RMS) of the accelerations over the frequency range from 1 to 1600 Hz. The results for the total accelerations are shown Table 9. All points show a significant reduction in overall acceleration.

Table 9 Acceleration totals of measurement points

Point	Original Acceleration Total [m/s^2]	Isolated Acceleration Total [m/s^2]	% Reduction
22	0.83	0.06	93
23	0.64	0.05	92
24	0.44	0.04	91
25	0.94	0.16	83
26	0.90	0.11	88
27	1.16	0.13	89
28	1.18	0.07	94
29	1.98	0.10	95
61	1.47	0.18	88
63	1.10	0.11	90
64	2.08	0.18	91

4.4 Conclusion

Isolating the gearbox guard from the base significantly reduces the vibration of the gearbox guard and the sound power emitted from the gearbox guard. The sound power of the decanter is increased by 0.7 dB due to isolating the gearbox guard. The increase in sound power is likely due to the energy that was used to vibrate the gearbox guard now being used to increase the vibration of the base and hopper.

Key finding from the testing were:

- The rotating assembly was the main source of vibrations within the decanter.
- Isolating components from the base was effective in reducing the sound power produced in that component.
- The vibrational energy generated from the rotating assembly must be dissipated to prevent it from increasing the sound power of components that are not isolated from it.

5 Modification – New Base

The decanter was modified by changing the cast iron base for a polymer concrete base. The new base was designed to have the same shape as the original. The new base had a composition of small stones, big stones, silica sand, and epoxy resin, in the weight ratios of 100:100:100:7. The new base material was selected due the higher damping properties of the polymer material compared to the original cast iron.

Figure 53 illustrates the construction of the new polymer base with the semitransparent blue region denoting the epoxy-gravel region and the metal insert shown in gray. The top mounting surfaces are for the main bearings and hopper mounts. All additional components were mounted to the new base in the same manner as they were mounted to the original base.

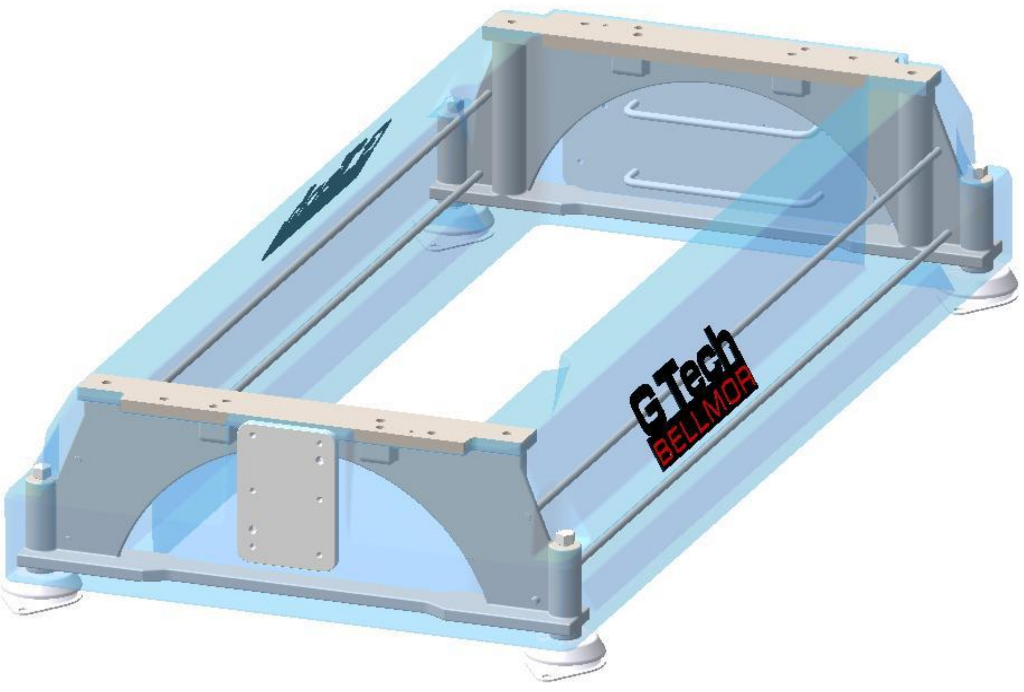


Figure 53 Polymer concrete base illustration

5.1 Results

The natural frequencies of the new base were determined by taking three measurement points along the side of the base and exciting the base with a mallet. The results of the measurements are shown in Figure 54, with the x-axis labelled at harmonics of the bowl rotation speed. The new base has a natural frequency near the bowl harmonic of 108 Hz and the base would be expected to vibrate at this frequency when the decanter is running.

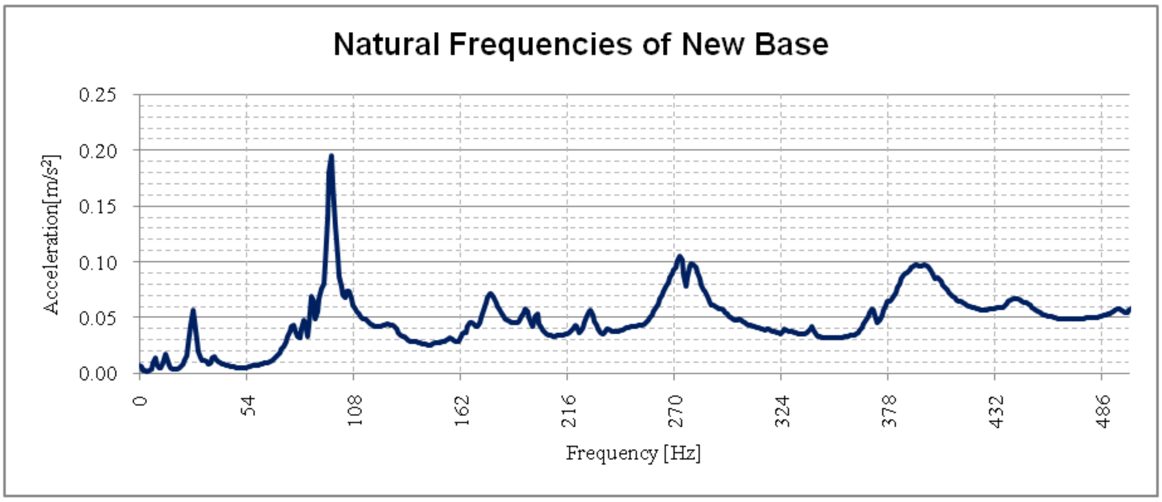


Figure 54 Natural frequency of the new base

The results of four sound intensity scans are shown in Figure 55. The scans were undertaken by the method described in Section 2.7. Four scans were done with the new base due to inconsistent results below the 500 Hz octave band. Adjustments were made to the tension of the main drive belts after the second scan. Scans three and four were deemed to be consistent as there was less than 0.6 dB difference in levels for one-third octave bands 80 to 4,000 Hz and were used for all subsequent comparisons. The sound power of the decanter was 106.1 dB

Vibration acceleration measurements were then taken after intensity scan four. The overall results for the vibration acceleration are shown in Table 10 and Figure 56. The overall accelerations had decreased but there was a significant increase in accelerations measured at 162 Hz, the third harmonic of the bowl speed.

Table 10 Overall acceleration results

	Averaged RMS Accelerations [m/s^2]			
Configuration	Base	Hopper	GB Guard	Total
Original	0.15	0.46	1.26	0.70
New Base	0.11	0.25	0.68	0.37

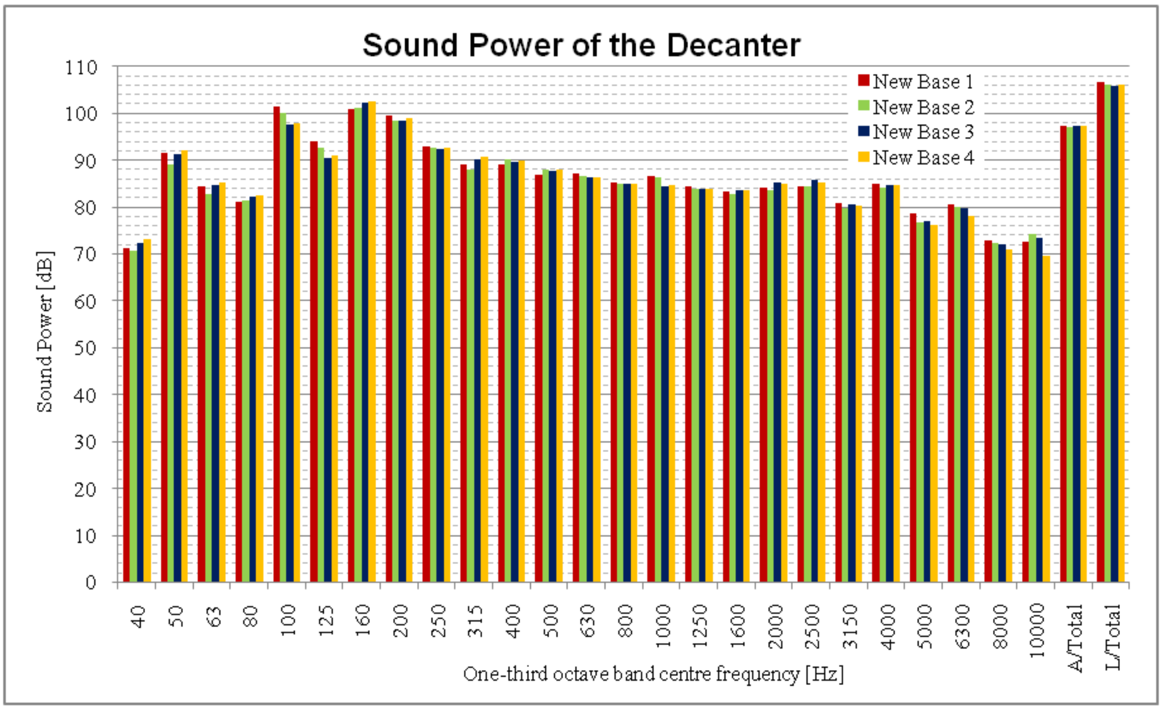


Figure 55 Sound power of the decanter

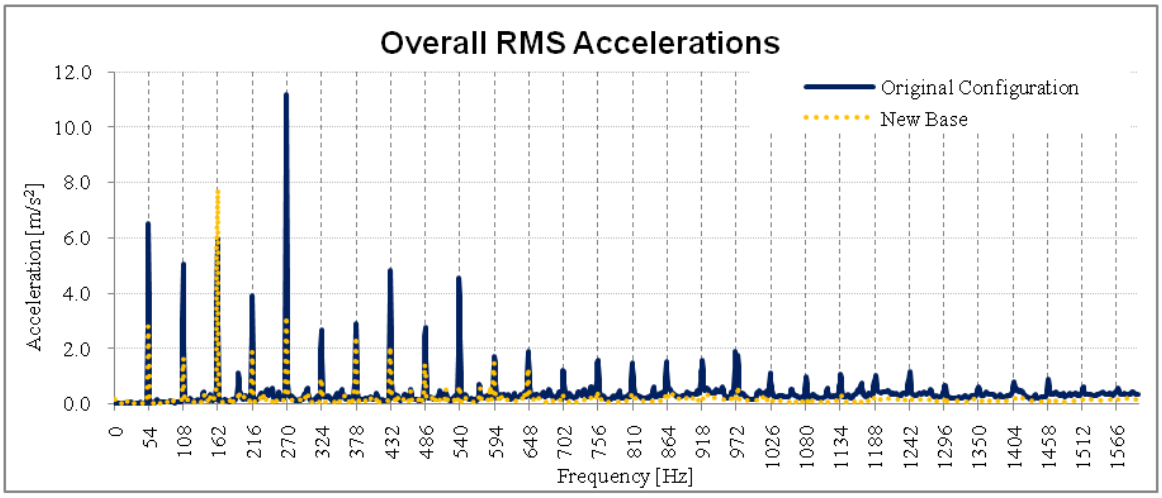


Figure 56 Total averaged RMS accelerations of driven harmonic frequencies

5.2 Comparison of Sound Intensity Measurements

A comparison of the sound powers of the decanter with the new base and with the original configuration, see Section 3.1.3, is shown in Figure 57. The comparison shows that the decanter produced significantly more sound power in the 80, 160, and 200 Hz one-third octave bands, with reductions in sound power in the 50, 63, and 400 Hz one-third octave bands. The overall effect was an increase in sound power of 2 dB. There was essentially no change to the dB(A) level, indicating that the perceived level for a human listener had not changed.

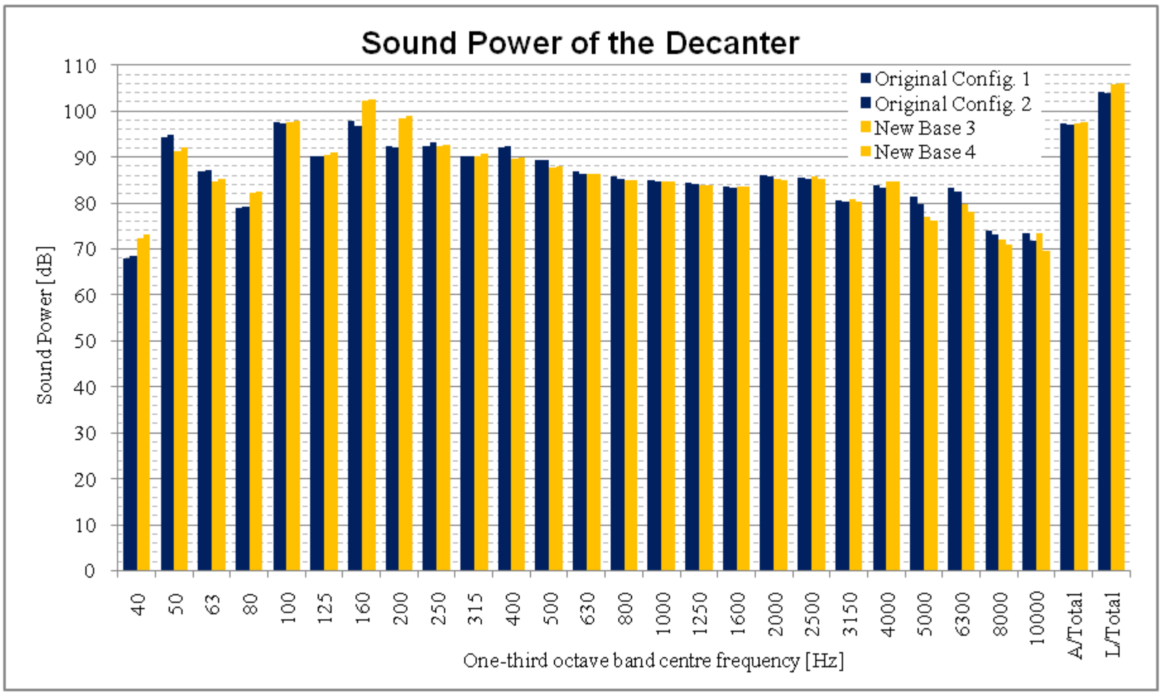


Figure 57 Sound power comparison

Tables 11 to 13 show the sound intensity for each region of the intensity scan with comparison to the original configuration. The scans are:

- OC1 – Original Configuration Scan 1
- OC2 – Original Configuration Scan 2
- NB3 – New Base Configuration Scan 3
- NB4 – New Base Configuration Scan 4

If the value in the table is 0.0, then this indicates that a negative intensity was measured and infers that there was a higher noise source behind the intensity probe for that frequency. This is primarily due to the large number of reflective surfaces within the testing room. These results have not been used to calculate the average difference in sound power of the two configuration comparison. The negative intensity measurements only occur in the region/frequency combination that had sound power measurements below 80 dB and do not influence the overall results, which are over 90 dB.

The scan regions that showed the most consistent increase in total sound intensity were the left side and lower regions of the front and back. The main contributor to the overall sound power was the 160 Hz one-third octave band. Table 12 shows the changes in sound intensity for this one-third octave band. It shows a significant increase in sound power for the left side and also significant increases in regions of the front, back and top which are closest to the left side (there was also a significant increase in the right side but the levels were still low relative to other regions). The results show that the gearbox guard was most likely the main contributor to the overall increase in sound power of the decanter.

The 200 Hz one-third octave band was the other significant contributor to the overall sound power and also significantly increased in level from the original configuration. Table 13 shows the sound intensity measurements for this one-third octave band. The front, back, left and top regions all have sound intensities between 84.5 dB and 88.3 dB. This indicated that all the major components under investigation (gearbox guard, base, and hopper) could be contributing to the increase in sound power for this octave band.

The two regions with the highest overall sound intensity, over 98 dB, were Left 2 and Top 1. These two regions are directly related to the gearbox guard. This indicated that the gearbox guard was the most significant source of sound power.

Table 11 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	92.7	91.4	91.9	87.9		89.4	91.5	93.0	87.1	91.1
OC2	93.1	91.6	93.0	88.2		89.4	91.7	93.1	87.4	91.5
NB3	93.0	91.2	92.6	85.9		94.5	94.7	94.9	90.1	92.9
NB4	92.2	91.1	92.6	84.6		95.7	95.7	95.2	90.7	93.4
Ave Diff.	→ -0.3	→ -0.3	→ 0.1	↓ -2.8		↑ 5.7	↑ 3.6	↑ 2.0	↑ 3.2	↑ 1.8
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	86.7	92.5	93.0	96.0		75.9	91.0	91.2	91.0	91.4
OC2	87.4	92.5	92.9	95.1		80.7	91.7	91.4	90.3	91.4
NB3	77.4	91.7	94.2	98.8		84.8	95.5	94.3	93.9	93.7
NB4	64.5	91.6	94.6	98.5		85.4	95.6	94.8	94.3	93.9
Ave Diff.	↓ -16.1	→ -0.8	↑ 1.5	↑ 3.1		↑ 6.8	↑ 4.2	↑ 3.2	↑ 3.4	↑ 2.4
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	93.4	97.8	96.2	91.9	94.6			89.9	90.5	90.2
OC2	93.8	97.0	95.5	92.0	94.2			88.9	89.6	89.3
NB3	97.7	101.0	97.0	95.3	97.7			87.3	89.6	88.7
NB4	98.3	101.7	96.9	95.7	98.2			86.9	89.5	88.5
Ave Diff.	↑ 4.4	↑ 3.9	↑ 1.1	↑ 3.6	↑ 3.5			↓ -2.3	→ -0.5	↓ -1.2
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	94.5	91.3	93.9	90.2	92.6					91.9
OC2	93.7	92.8	92.1	88.7	92.0					91.8
NB3	98.5	94.0	89.7	87.6	93.6					93.8
NB4	98.5	93.8	89.7	87.7	93.5					94.1
Ave Diff.	↑ 4.4	↑ 1.9	↓ -3.3	↓ -1.8	↑ 1.2					↑ 2.1

Table 12 Comparison of sound intensity measurements for the 160 Hz one-third octave band [dB]

One-third octave band centre frequency:				160	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	89.0	80.0	78.0	83.2		82.5	74.8	84.7	82.6	82.9
OC2	89.5	81.5	75.8	82.3		83.7	72.6	83.0	79.1	82.4
NB3	91.1	69.7	77.6	74.5		86.7	82.0	88.1	86.9	85.3
NB4	89.4	0.0	74.9	0.0		85.5	84.2	87.1	86.6	84.4
Ave Diff.	1.0	-11.1	-0.7	-8.3		3.0	9.4	3.8	5.9	2.2
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	0.0	79.2	87.0	94.4		0.0	0.0	81.2	85.6	84.6
OC2	0.0	76.5	85.8	93.2		0.0	76.1	75.3	84.6	83.6
NB3	0.0	0.0	91.2	98.2		69.7	80.8	0.0	91.6	88.8
NB4	0.0	0.0	91.3	97.9		77.5	80.0	78.8	91.9	88.8
Ave Diff.	-	-	4.8	4.3		-	4.3	0.5	6.7	4.7
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	91.9	96.1	92.2	86.7	91.8			0.0	71.2	0.0
OC2	91.6	94.9	90.8	87.1	91.0			0.0	76.8	66.1
NB3	97.1	100.4	95.9	92.2	96.4			0.0	78.0	74.0
NB4	97.8	101.2	95.6	92.8	97.0			67.1	77.4	75.0
Ave Diff.	5.7	5.3	4.3	5.6	5.3			-	3.7	8.4
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	90.1	71.0	88.0	0.0	85.3					85.5
OC2	90.3	82.2	71.6	0.0	83.4					84.6
NB3	97.7	91.7	80.3	78.3	91.5					90.1
NB4	97.6	90.9	81.0	78.8	91.2					90.2
Ave Diff.	7.5	14.7	0.9	-	7.0					5.1

Table 13 Comparison of sound intensity measurements for the 200 Hz one-third octave band [dB]

One-third octave band centre frequency:				200	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	82.0	80.5	79.8	76.2		81.5	79.8	79.9	75.9	79.7
OC2	82.5	81.1	78.7	75.4		82.9	81.6	81.6	76.4	80.5
NB3	82.0	81.6	82.4	0.0		91.6	91.0	86.6	79.4	86.6
NB4	81.7	82.7	83.3	0.0		92.7	91.7	86.9	81.4	87.4
Ave Diff.	-0.4	1.4	3.6	-		10.0	10.7	6.0	4.3	6.9
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	73.0	78.2	79.5	82.3		0.0	82.9	82.1	83.8	80.2
OC2	71.1	78.9	77.9	81.2		0.0	82.3	81.3	80.7	79.3
NB3	72.2	87.8	86.8	84.3		0.0	90.7	88.7	82.5	86.7
NB4	73.5	87.9	87.2	85.0		0.0	91.4	88.8	82.0	87.2
Ave Diff.	0.8	9.3	8.3	2.9		-	8.5	7.1	0.0	7.2
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	79.1	82.0	81.3	80.5	80.8			67.8	80.7	78.2
OC2	79.5	81.9	81.3	79.5	80.4			66.5	79.3	76.8
NB3	81.9	85.2	87.2	89.5	88.0			78.5	81.4	80.3
NB4	83.6	86.4	87.8	89.5	88.3			78.9	81.3	80.4
Ave Diff.	3.5	3.9	6.2	9.5	7.6			11.6	1.3	2.8
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	82.9	78.9	82.2	78.4	80.8					80.1
OC2	81.6	82.8	78.4	78.0	80.6					79.9
NB3	87.7	86.0	80.9	81.2	84.5					86.1
NB4	88.7	87.2	80.9	81.2	85.3					86.7
Ave Diff.	6.0	5.8	0.6	3.0	4.2					6.4

5.3 Comparison of Vibration Acceleration Measurements

Figures 58 to 66 show the vibration acceleration measurement results of the three main components of the decanter under investigation. The acceleration measurements have been combined by taking the 'root of the mean of the squares' (RMS) of the various individual measurements.

The accelerations measured within the base decreased except for the following instances: accelerations at 108 Hz, vertical accelerations at 216 and 432 Hz. There was no increase in sound power for the 100 Hz one-third octave band relating to the increase in acceleration at 108 Hz. This can be explained by the decrease in acceleration measured at 108 Hz for the gearbox guard, and hopper.

The increase in vibration measured at 216 Hz was expected as it directly relates to the increase in sound power for the 200 Hz one-third octave band. The 400 Hz one-third octave band had a decrease in sound power for the decanter. The increase in vibration measured in the base at 432 Hz was not high enough to offset the decrease at this frequency for the other components.

The vibration of the gearbox guard also decreased, except at 162 Hz and laterally at 378 Hz. The significant increase in accelerations at 162 Hz support the earlier determination that the gearbox was the likely the main source for the increase in sound power at the 160 Hz one-third octave band. There was no increase in sound power for the one-third octave band that relates to the 378 Hz acceleration increase. This was likely due to the other components having decreases at this at this frequency.

The hopper had similar decreases in acceleration measurements except for 162 Hz, laterally at 216 Hz. This indicated that the hopper was a likely contributor to the increase in sound power measured in the 160 and 200 Hz one-third octave bands.

All three components had a decrease in the overall vibration measured as shown in Table 10 and Figures 58 to 66. This decrease in overall vibration has not resulted in a decrease in the sound power produced by the decanter. Only changes in the higher vibration levels have influenced the overall sound power produced by the decanter. Reducing the lower vibration levels did not significantly alter the overall sound power of the decanter.

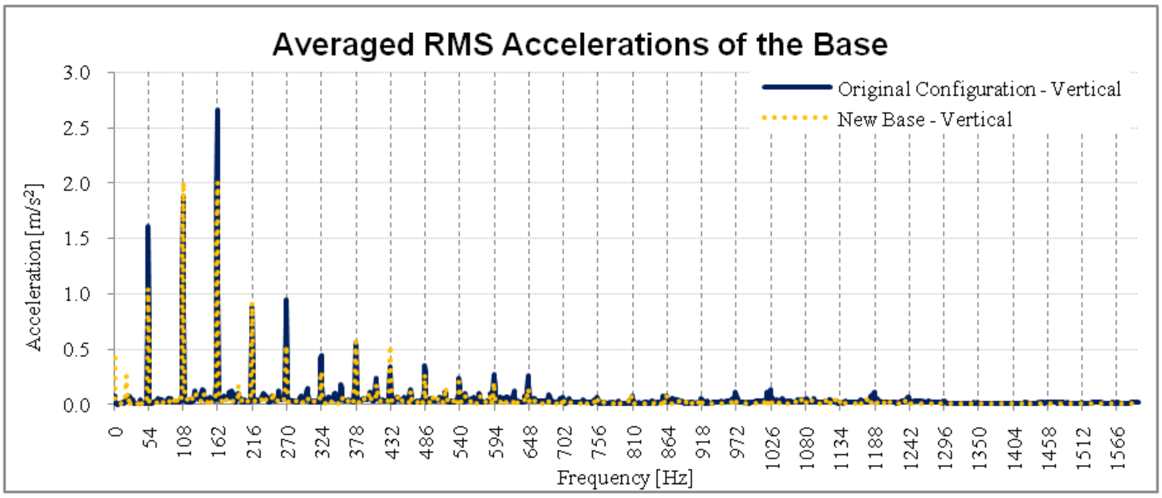


Figure 58 Accelerations of the base – vertical

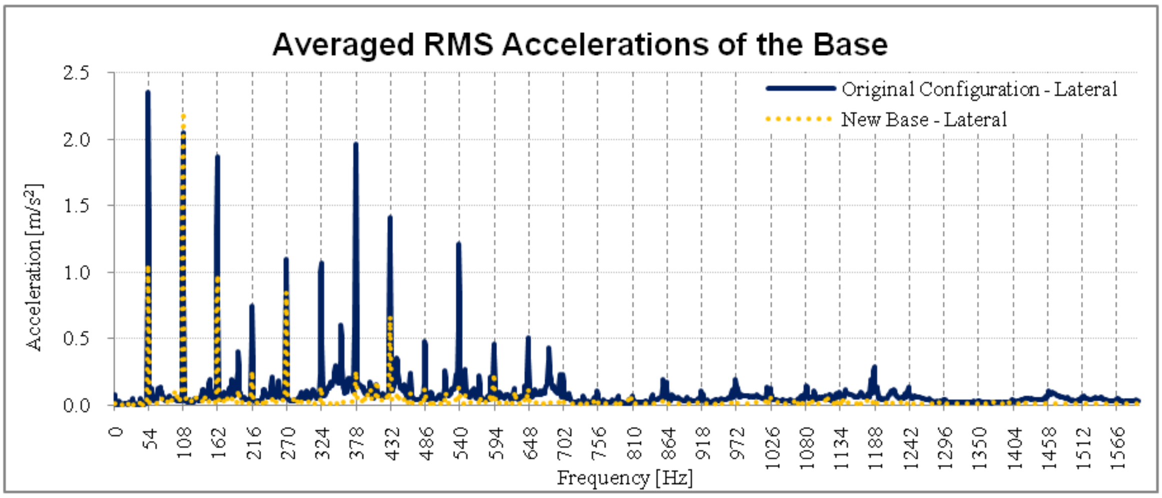


Figure 59 Accelerations of the base – lateral

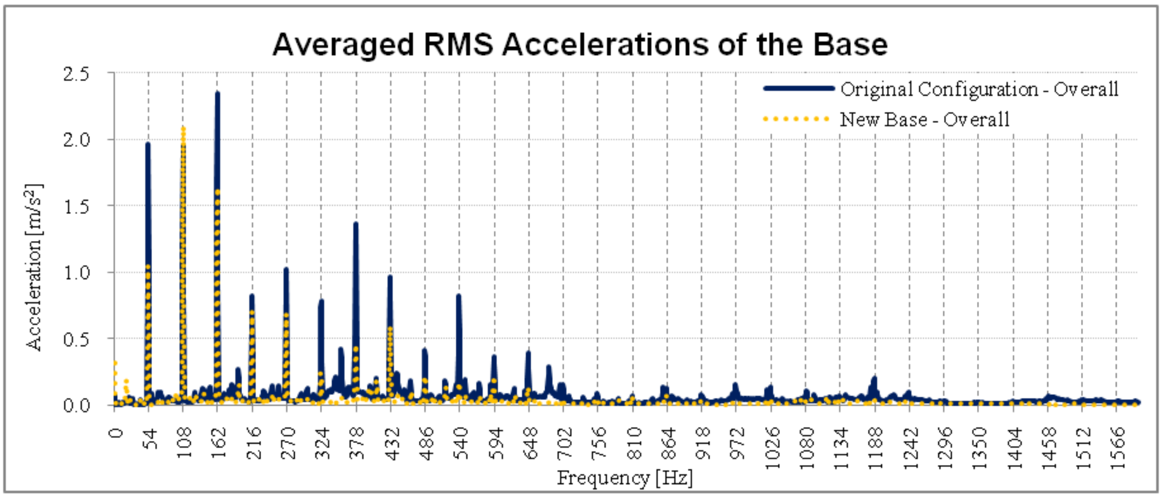


Figure 60 Accelerations of the base – overall

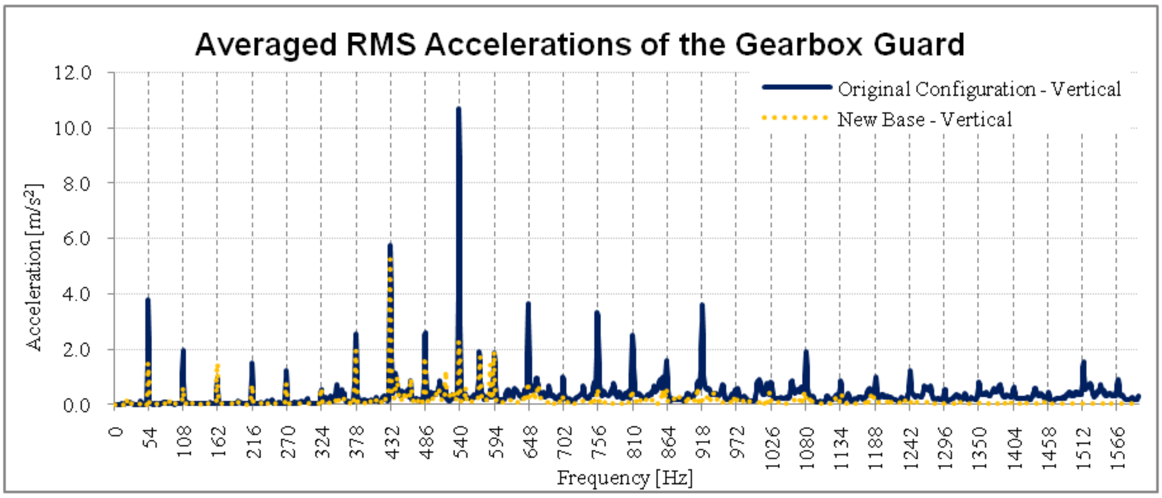


Figure 61 Accelerations of the gearbox guard – vertical

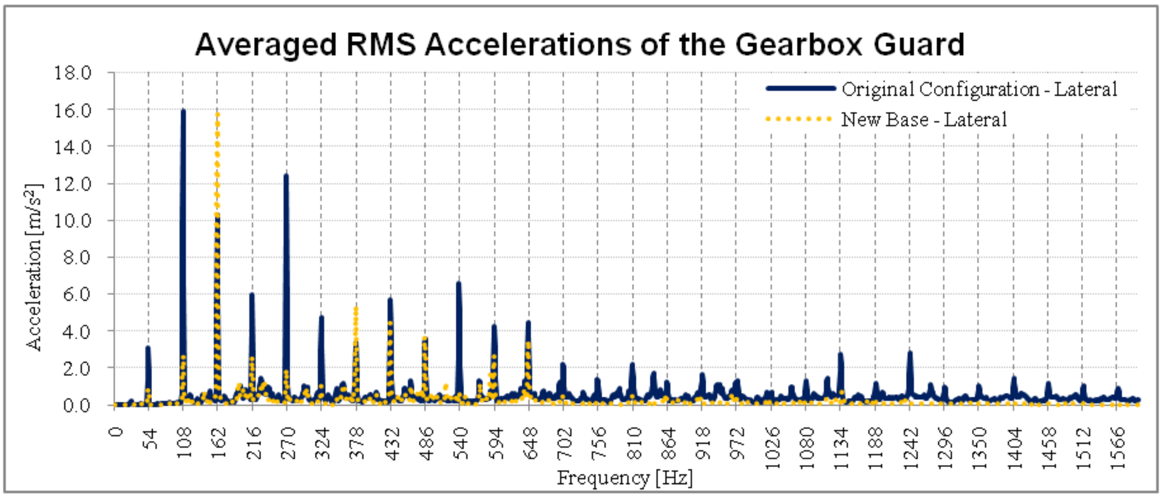


Figure 62 Accelerations of the gearbox guard – lateral

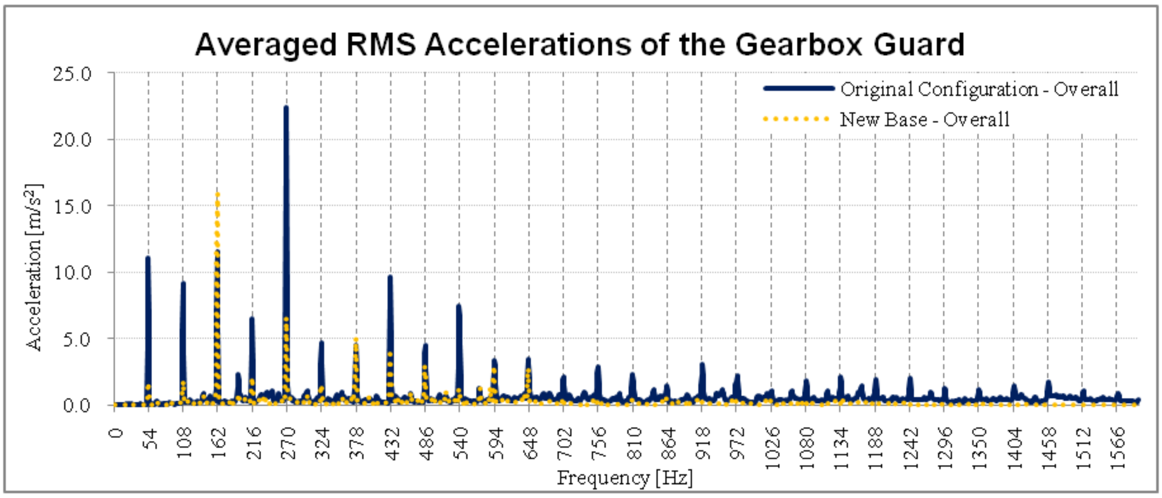


Figure 63 Accelerations of the gearbox guard – overall

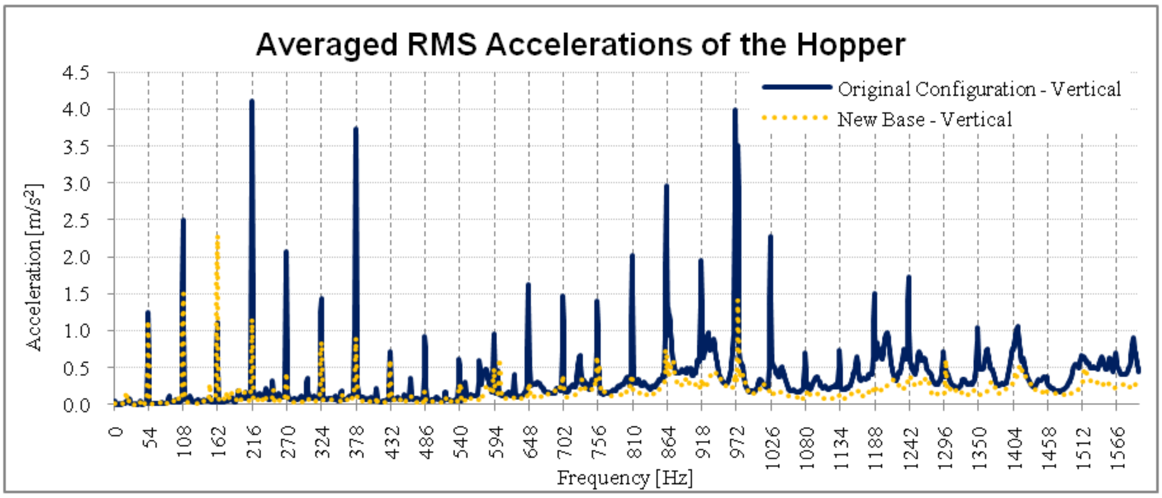


Figure 64 Accelerations of the hopper – vertical

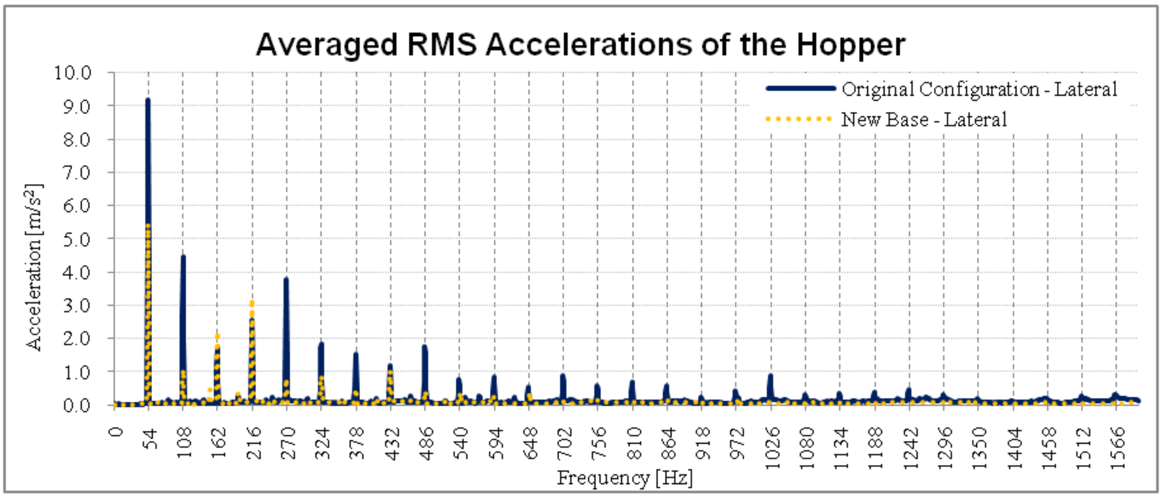


Figure 65 Accelerations of the hopper – lateral

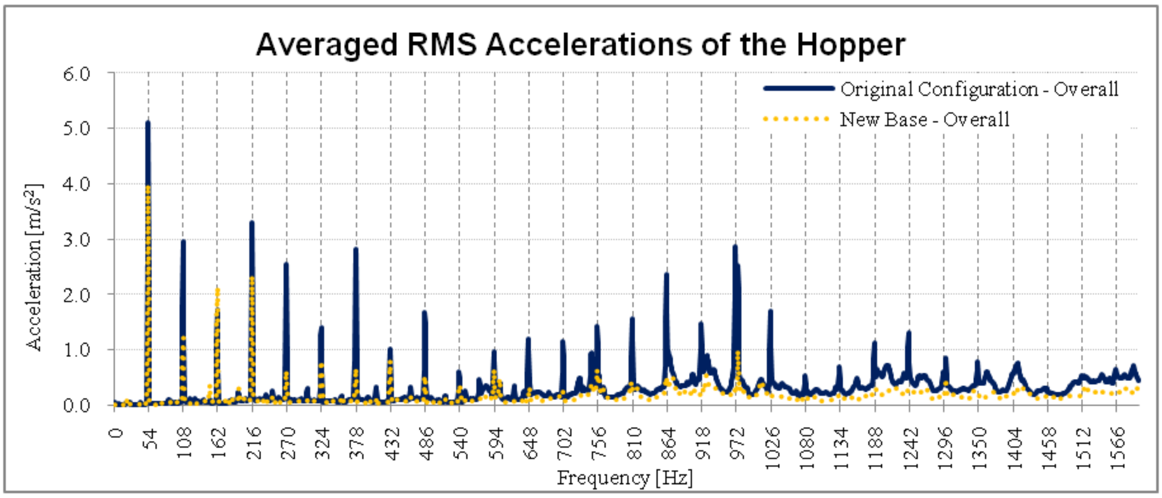


Figure 66 Accelerations of the hopper – overall

5.4 Discussion

It is clear that the modification of the decanter by using a polymer concrete base has nearly halved the vibration accelerations for all the components measured. The decrease in accelerations has generally occurred across the entire spectrum, the exceptions being at the crucial harmonic frequencies of 108, 162, and 216 Hz. The 100, 160, and 200 Hz one-third octave bands are the bands that contain the three crucial harmonic frequencies and are also the bands that have the highest sound power levels. The new base also resulted in the various components vibrating less at the bowl speed, 54 Hz, but more at one, or more, of the second, third or fourth harmonics of the bowl speed.

The changes in the vibrating patterns of the various components have had the following effects on the sound power produced by the decanter:

- A 2.7 dB decrease in the 50 Hz one-third octave band.
- A 0.3 dB increase in the 100 Hz one-third octave band.
- A 5.1 dB increase in the 160 Hz one-third octave band.
- A 6.4 dB increase in the 200 Hz one-third octave band.
- A 2 dB increase in overall sound power of the decanter.
- No change to the overall sound power level (A-weighted).

All three components showed significant reductions in acceleration measurements above 500 Hz but there was no reduction in sound power measurements from the intensity scans. This indicates that the vibrations of these components were not likely the main source of noise for these higher frequencies.

To reduce the increases in vibration of the various components they need to be isolated from the vibration source or redesigned to alter their natural vibration pattern. The vibration source is from the spinning bowl which is attached to the base via two main bearings. It is not possible to isolate the base from the vibration source, hence the design and properties of the materials used to construct the base need investigation to reduce the noise generated by it. As the gearbox guard is the most significant source of sound power it should be the first component investigated to reduce the overall sound power of the decanter.

The gearbox guard was isolated from the original base with mixed success. The isolated gearbox guard had significantly reduced acceleration measurements and the sound intensity radiated from it was also reduced. The overall sound power did increase due to the rest of the decanter producing more noise. With the improved damping characteristics of the new polymer concrete base, isolating the gearbox guard should prove more effective at lowering the overall sound power of the decanter than isolating the gearbox guard from the original base, see Section 4. Changing the material properties of the gearbox guard has been investigated and found that ultra high molecular weight polyethylene (UHMWPE) plastic had higher predicted radiated acoustic power than mild steel (current material) for frequencies below 378 Hz [14]. As most of the sound power for the decanter is below 378 Hz, different materials for guards will not be investigated. Isolation of the hopper assembly from the base is expected to achieve similar results to isolating the gearbox guard.

5.5 Conclusion

The decanter modification of using a polymer concrete base resulted in a 2 dB increase in sound power. The increase in sound power was due to increases within the 160 and 200 Hz one-third octave bands and the main contributors to the increase were the gearbox guard and hopper. The increase in sound power in the 200 Hz one-third octave band was due to the base and hopper. An increase in vibration at 108 Hz in the base also ensured that the sound power in the 100 Hz one-third octave band did not decrease. There was a significant reduction in the measured vibration levels over 500 Hz for the base, gearbox guard and hopper with corresponding reduction in sound power over 500 Hz. This indicates that the main sound source over 500 Hz is not due to the vibration of the base, gearbox guard or hopper.

The use of a polymer concrete base resulted in the following changes in sound power:

- A decrease in sound power relating to the bowl speed (54 Hz – 50 Hz one-third octave band)
- No change in sound power relating to the second harmonic (108 Hz – 100 Hz one-third octave band)
- An increase in sound power relating to the third and fourth harmonic (162, 216-160, 200 Hz one-third octave band)

- No changes in sound power for the 630 to 4,000 Hz one-third octave bands

It is recommended that isolation of the gearbox guard and hopper from the new base be investigated as possible measures to reduce the sound power of the decanter.

6 Modification – New Base with Gearbox Guard Isolation

The gearbox guard was isolated from the new polymer concrete base by mounting the gearbox guard to brackets that were bolted to the sub frame, as illustrated in Figure 67. The positioning of the gearbox guard remained the same except it was spaced approximately 15 mm off the base to make room for bolts holding the guard to the bracket. This was the same way that the gearbox guard was isolated for tests using the original base, see Section 4. This configuration will be referred to as NB-GI.

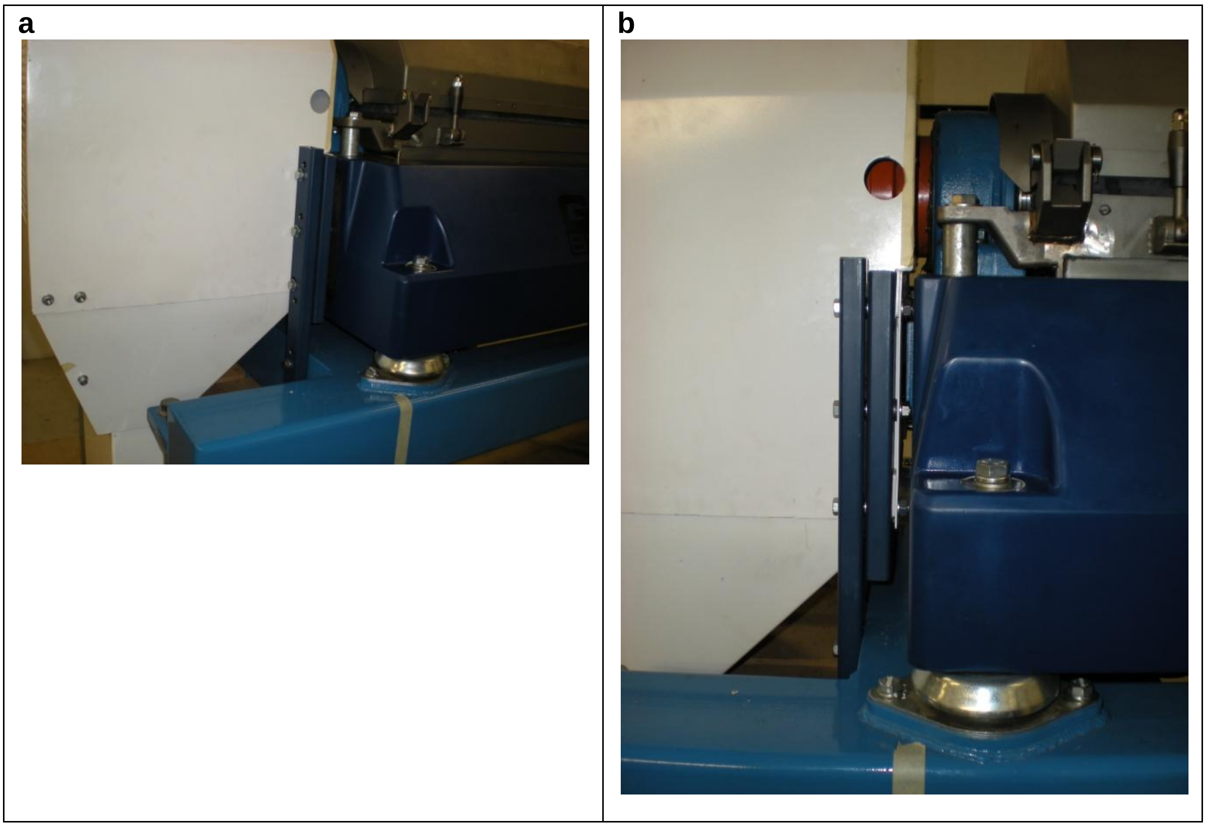


Figure 67 (a-b) Photographs of the isolation of the gearbox guard from the new base

6.1 Results

Results of the three sound intensity scans are shown Figure 68. The scans were undertaken by the method described in Section 2.7. The first two scans did not produce consistent results across the one-third octave bands and so a third scan was carried out. The second and third scans had less than 0.5 dB difference for all one-third octave bands above 63 Hz. The results from scans two and three were used in all subsequent analysis for this new configuration. The sound power of the decanter was 103.5 dB.

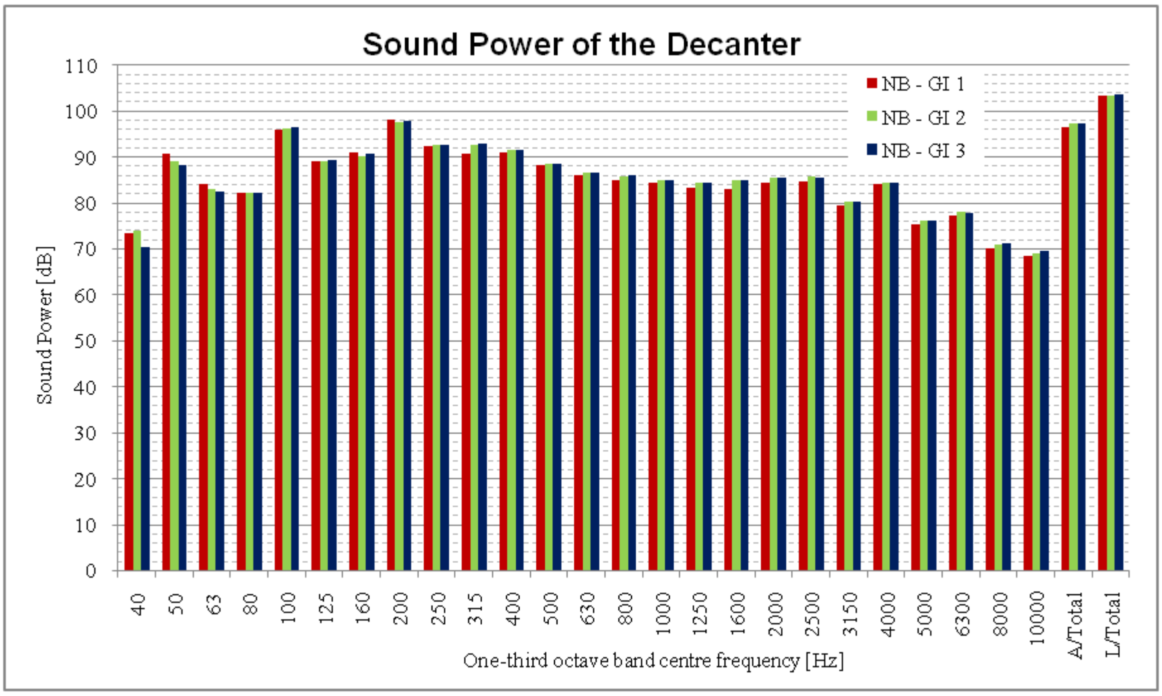


Figure 68 Sound power of the decanter

Vibration acceleration measurements were taken after scan three was completed. The overall results for the acceleration measurements are shown in Table 14 and the spectrum is shown in Figure 69. The accelerations measured on the gearbox guard decreased to 17 % of their previous configuration, New Base, see Section 5. The accelerations for all the lower driven harmonic frequencies have reduced, particularly that of the third harmonic, 162 Hz.

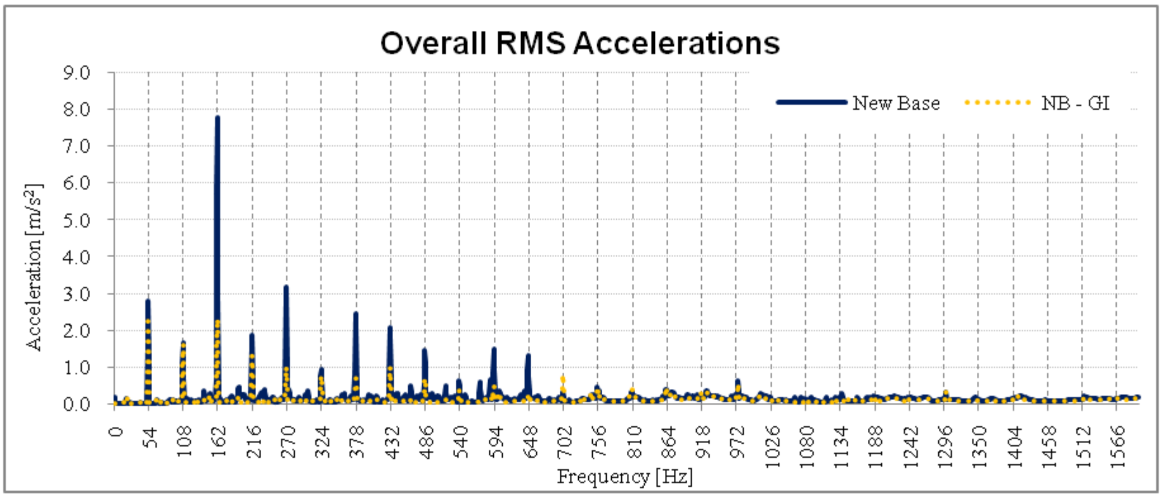


Figure 69 Total RMS average accelerations of driven harmonic frequencies

Table 14 Overall acceleration results

	RMS Average Accelerations [m/s^2]			
Configuration	Base	Hopper	GB Guard	Total
New Base	0.11	0.25	0.68	0.37
NB - GI	0.12	0.25	0.12	0.18

6.2 Comparisons

6.2.1 Sound Intensity Measurements

The comparison of the sound intensity scans for the current decanter configuration, NB-GI, and the previous configuration, New Base, are shown in Figure 70 and Tables 15 to 17. The most notable difference in the sound power measurements is the 12 dB reduction in the 160 Hz one-third octave band. The modification has reduced the sound power below the 250 Hz one-third octave band and slightly increased the sound power above the 250 Hz one-third octave band. The modification resulted in a 2.6 dB reduction in sound power produced by the decanter.

Table 15 shows that the major reduction in the overall sound intensity occurred at the left hand end of the decanter, with the left side and regions of the front, back and top. The 15.1 dB increase in region Back 1 is not significant, due to the previous very low levels emitted through this region and the subsequent levels still being at the lower end of levels measured. The increase can be attributed to how the sound field established itself for this new configuration (especially so, as amount and location of equipment at the other end of the test room may have changed).

The 160 Hz one-third octave band sound intensity results are shown in Table 16. It shows that all the regions directly associated with the gearbox guard have an average sound power reduction of over 17 dB. The total sound power reduction for this one-third octave band is 12 dB.

There are a number of 0.0 values within the table. This probably indicates that reflections from the surroundings have impacted on the intensity measurements for regions that have low emissions at this low frequency. The overall values are not affected as a decrease in one region due to a reflection will be matched by an increase in another region as the reflected wave passes back out of the virtual enclosure. This is why there are some increases in some regions for no apparent reason.

The largest increase in sound power was in the 315 Hz one-third octave band and the results of the sound intensity scans are shown in Table 17. There are consistent increases from most regions of the machine that cannot be attributed to a particular component of the decanter. This pattern is typical for the one-third octave bands above 250 Hz, little change or a slight decrease through the left side and a slight increase in the remaining sides.

For the new base configuration, the regions with the highest sound intensity were Left 2 and Top 1. These two regions are directly related to the gearbox guard. This indicates that the gearbox guard was the main source of sound power for the new base configuration of the decanter. As the regions directly associated with the gearbox guard, (Left 2, Top 1, Front 1, Back 4), are all now 90 dB or below in sound intensity, the gearbox guard is no longer the source with the highest sound intensity for the decanter in the NB-GI configuration.

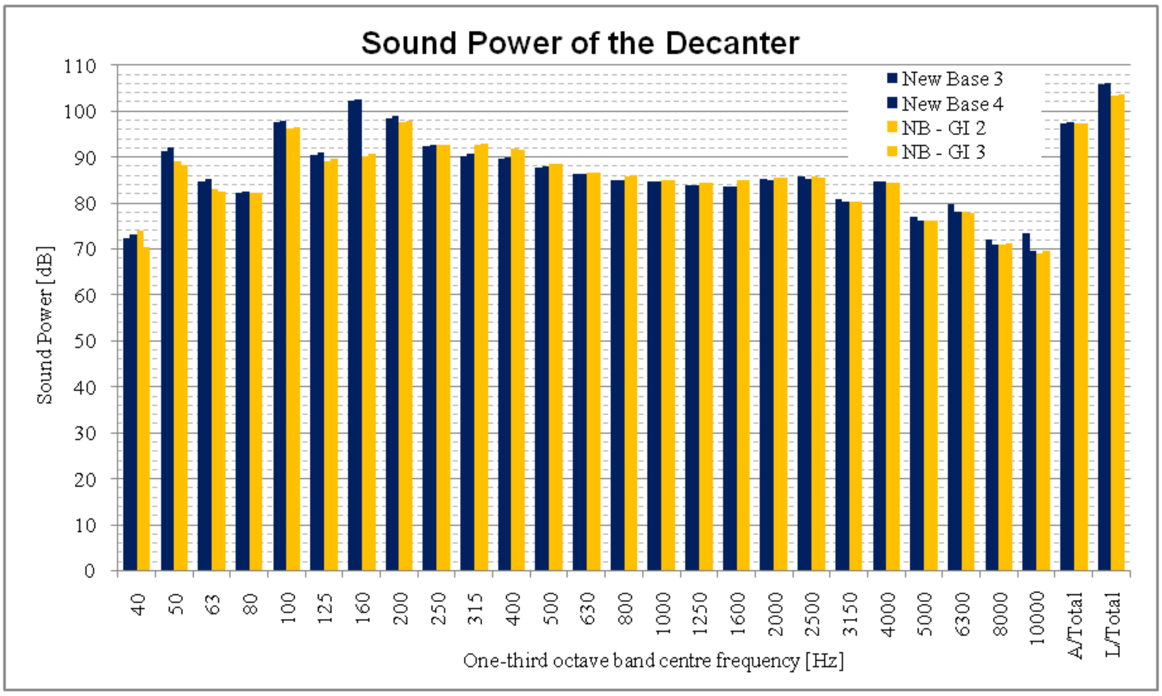


Figure 70 Sound power comparison

Table 15 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB3	93.0	91.2	92.6	85.9		94.5	94.7	94.9	90.1	92.9
NB4	92.2	91.1	92.6	84.6		95.7	95.7	95.2	90.7	93.4
NB-GI2	88.2	90.3	92.0	85.6		93.9	95.4	94.0	86.6	92.2
NB-GI3	87.5	90.0	93.2	87.2		93.8	94.8	94.3	87.8	92.4
Ave Diff.	-4.8	-1.0	0.0	1.2		-1.3	-0.1	-0.9	-3.2	-0.8
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB3	77.4	91.7	94.2	98.8		84.8	95.5	94.3	93.9	93.7
NB4	64.5	91.6	94.6	98.5		85.4	95.6	94.8	94.3	93.9
NB-GI2	85.8	91.2	90.3	88.6		85.7	95.7	94.2	89.1	91.9
NB-GI3	86.3	91.4	90.6	88.6		83.1	96.0	94.7	89.5	92.2
Ave Diff.	15.1	-0.3	-4.0	-10.1		-0.7	0.3	-0.1	-4.8	-1.7
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB3	97.7	101.0	97.0	95.3	97.7			87.3	89.6	88.7
NB4	98.3	101.7	96.9	95.7	98.2			86.9	89.5	88.5
NB-GI2	88.2	89.3	89.6	91.3	90.4			87.0	88.2	87.7
NB-GI3	87.5	88.8	89.5	91.7	90.6			87.1	89.3	88.4
Ave Diff.	-10.2	-12.3	-7.4	-4.0	-7.4			0.0	-0.8	-0.5
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB3	98.5	94.0	89.7	87.6	93.6					93.8
NB4	98.5	93.8	89.7	87.7	93.5					94.1
NB-GI2	90.0	90.7	89.9	86.9	89.6					91.2
NB-GI3	89.8	90.7	89.9	87.0	89.6					91.4
Ave Diff.	-8.6	-3.2	0.2	-0.7	-3.9					-2.6

Table 16 Comparison of sound intensity measurements for the 160 Hz one-third octave band [dB]

One-third octave band centre frequency:				160	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB3	91.1	69.7	77.6	74.5		86.7	82.0	88.1	86.9	85.3
NB4	89.4	0.0	74.9	0.0		85.5	84.2	87.1	86.6	84.4
NB-GI2	75.3	0.0	0.0	0.0		80.1	83.1	84.3	0.0	73.1
NB-GI3	77.0	0.0	0.0	0.0		79.2	82.0	84.3	0.0	75.5
Ave Diff.	-14.1	-	-	-		-6.4	-0.5	-3.3	-	-10.6
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB3	0.0	0.0	91.2	98.2		69.7	80.8	0.0	91.6	88.8
NB4	0.0	0.0	91.3	97.9		77.5	80.0	78.8	91.9	88.8
NB-GI2	0.0	0.0	0.0	77.8		0.0	83.9	82.1	79.2	78.2
NB-GI3	0.0	70.3	0.0	77.1		0.0	86.1	82.3	78.6	79.5
Ave Diff.	-	-	-	-20.6		-	4.6	3.4	-12.9	-9.9
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB3	97.1	100.4	95.9	92.2	96.4			0.0	78.0	74.0
NB4	97.8	101.2	95.6	92.8	97.0			67.1	77.4	75.0
NB-GI2	79.6	77.3	77.5	80.6	79.6			74.7	79.1	77.6
NB-GI3	78.2	78.3	76.2	80.1	79.2			75.4	79.7	78.2
Ave Diff.	-18.6	-23.0	-18.9	-12.2	-17.3			8.0	1.7	3.4
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB3	97.7	91.7	80.3	78.3	91.5					90.1
NB4	97.6	90.9	81.0	78.8	91.2					90.2
NB-GI2	78.7	82.0	84.5	72.1	81.7					78.4
NB-GI3	79.5	81.5	82.8	73.6	80.7					78.8
Ave Diff.	-18.6	-9.6	3.0	-5.7	-10.2					-11.6

Table 17 Comparison of sound intensity measurements for the 315 Hz one-third octave band [dB]

One-third octave band centre frequency:				315	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB3	75.1	77.8	80.6	76.5		76.4	80.4	83.2	75.7	79.6
NB4	76.7	79.4	81.5	77.4		76.4	80.1	83.8	77.6	80.3
NB-GI2	75.5	80.4	83.1	79.2		77.1	81.4	85.6	80.9	81.9
NB-GI3	73.5	79.8	82.5	79.1		77.4	81.3	86.3	81.9	82.1
Ave Diff.	⇩ -1.4	⇧ 1.5	⇧ 1.8	⇧ 2.2		⇨ 0.8	⇧ 1.1	⇧ 2.4	⇧ 4.8	⇧ 2.1
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB3	73.0	78.9	76.3	74.7		71.8	82.5	80.3	76.4	78.6
NB4	73.6	78.7	76.4	73.9		75.0	82.1	80.6	76.2	78.6
NB-GI2	79.6	83.4	77.7	77.7		72.3	85.7	80.6	78.7	81.4
NB-GI3	80.1	84.3	78.3	78.1		68.4	86.1	81.6	79.6	82.0
Ave Diff.	⇧ 6.6	⇧ 5.0	⇧ 1.7	⇧ 3.6		⇩ -3.1	⇧ 3.6	⇨ 0.7	⇧ 2.8	⇧ 3.1
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB3	73.6	75.8	75.2	76.7	76.0			74.3	78.4	77.0
NB4	74.0	75.7	75.8	76.7	76.1			74.1	79.1	77.4
NB-GI2	75.5	77.6	76.7	75.9	76.4			73.9	80.5	78.6
NB-GI3	76.5	76.8	76.0	76.6	76.6			75.5	80.1	78.5
Ave Diff.	⇧ 2.2	⇧ 1.4	⇨ 0.8	⇨ -0.5	⇨ 0.4			⇨ 0.5	⇧ 1.6	⇧ 1.3
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB3	72.6	74.0	75.3	74.9	74.5					78.1
NB4	73.8	75.2	75.5	75.1	75.1					78.5
NB-GI2	72.4	76.2	77.7	76.0	76.3					80.4
NB-GI3	72.5	76.5	77.3	75.7	76.2					80.7
Ave Diff.	⇨ -0.7	⇧ 1.8	⇧ 2.1	⇨ 0.8	⇧ 1.4					⇧ 2.3

The regions with the highest sound intensity, over 94 dB, in the NB-GI configuration were Front 6 and 7 and Back 6 and 7. These regions are not directly associated to a component but it is likely that the noise emission is from the hopper, base or back-drive motor.

6.2.2 Acceleration Measurements

Figures 71 to 79 show the vibration acceleration measurements of the three main components under investigation on the decanter. The acceleration measurements have been combined by taking the ‘root of the mean of the squares’ (RMS) of the various individual measurements.

The accelerations measured on the base have generally increased for the harmonic frequencies with the remaining frequencies unchanged. The accelerations on the gearbox guard all decreased except for the lateral 54 Hz measurement. The hopper accelerations for the 54, 180 and 216 Hz harmonics have had little change or decreased. The remaining harmonic frequencies showed a general increase in accelerations measured with the intermediate frequency levels unchanged.

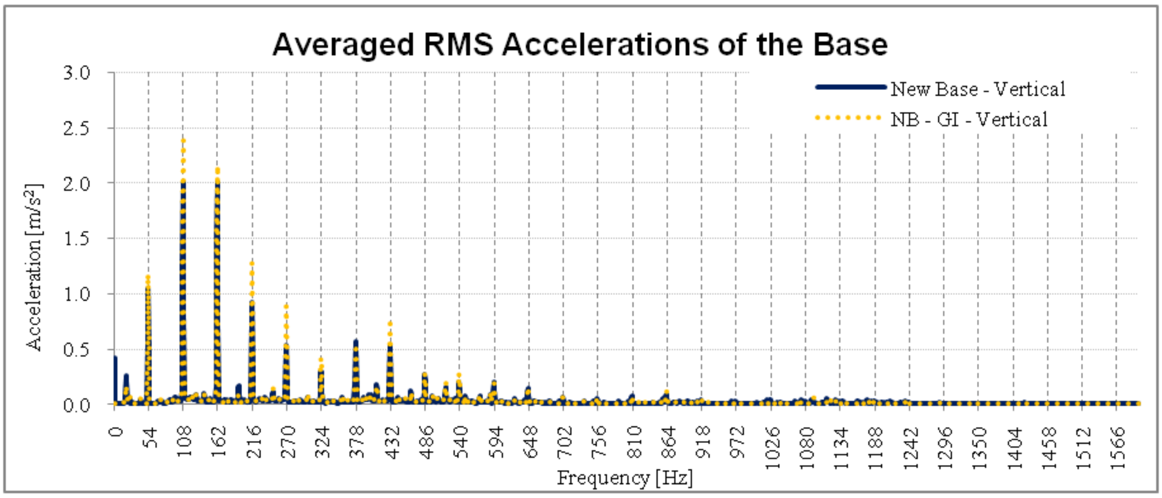


Figure 71 Accelerations of the base – vertical

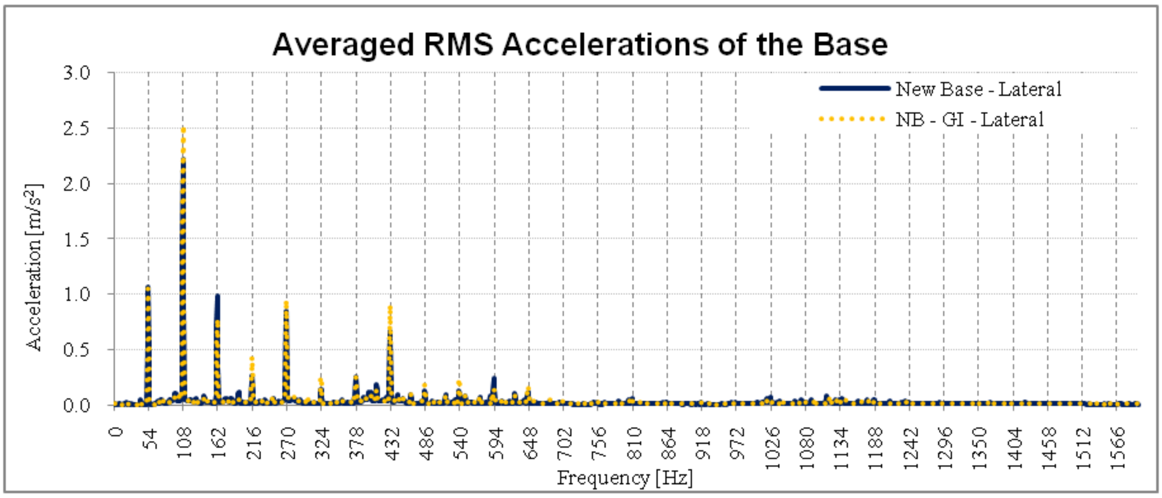


Figure 72 Accelerations of the base – lateral

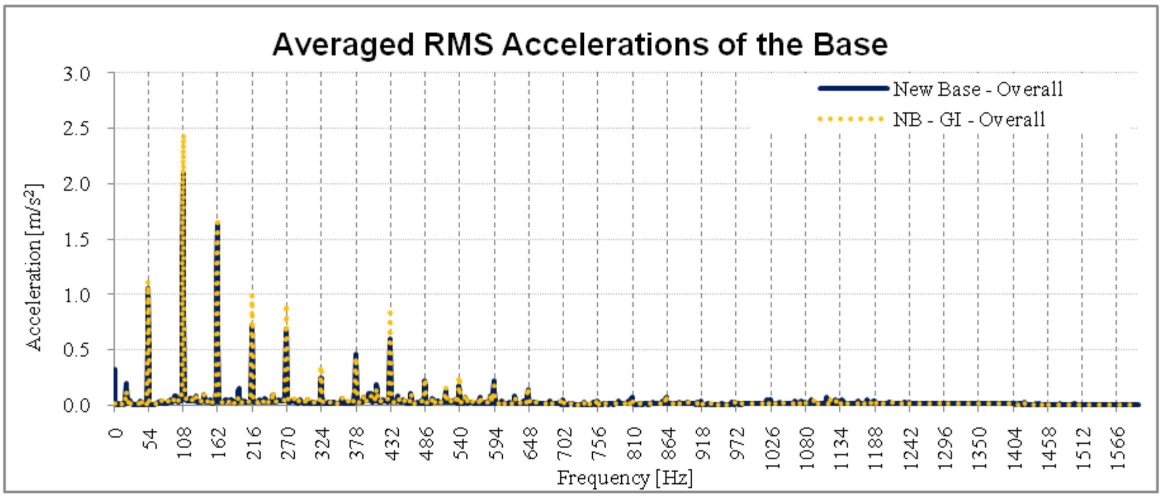


Figure 73 Accelerations of the base – overall

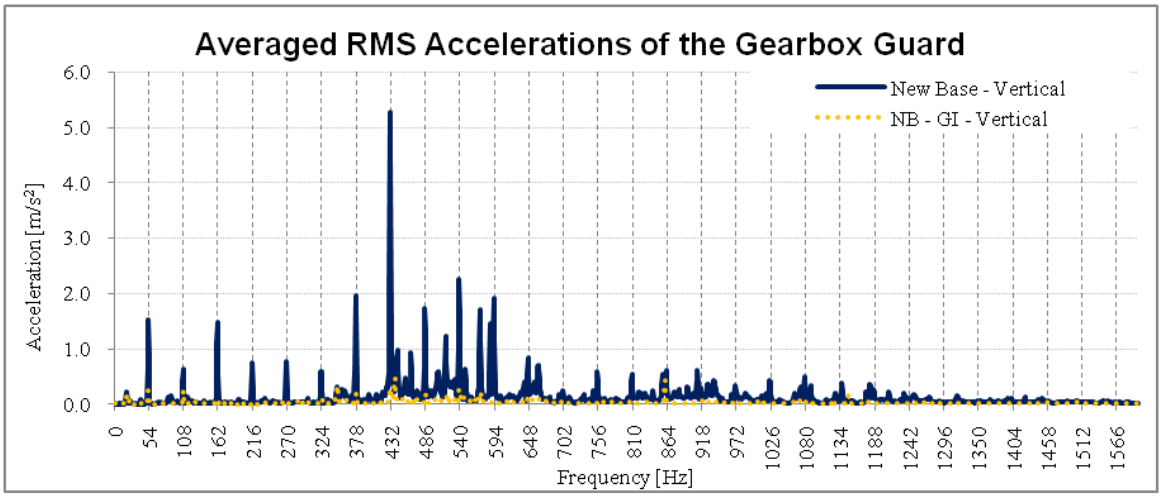


Figure 74 Accelerations of the gearbox guard – vertical

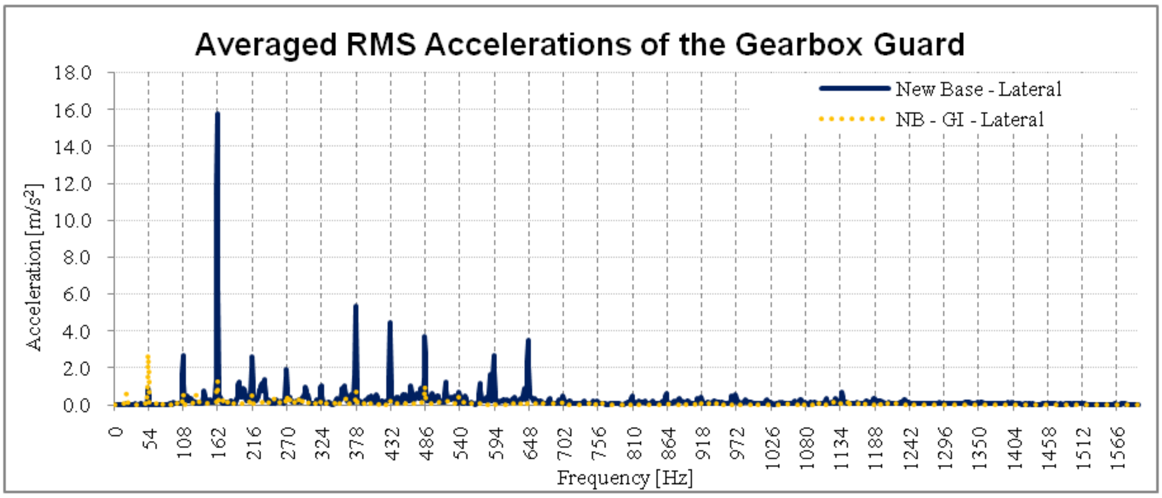


Figure 75 Accelerations of the gearbox guard – lateral

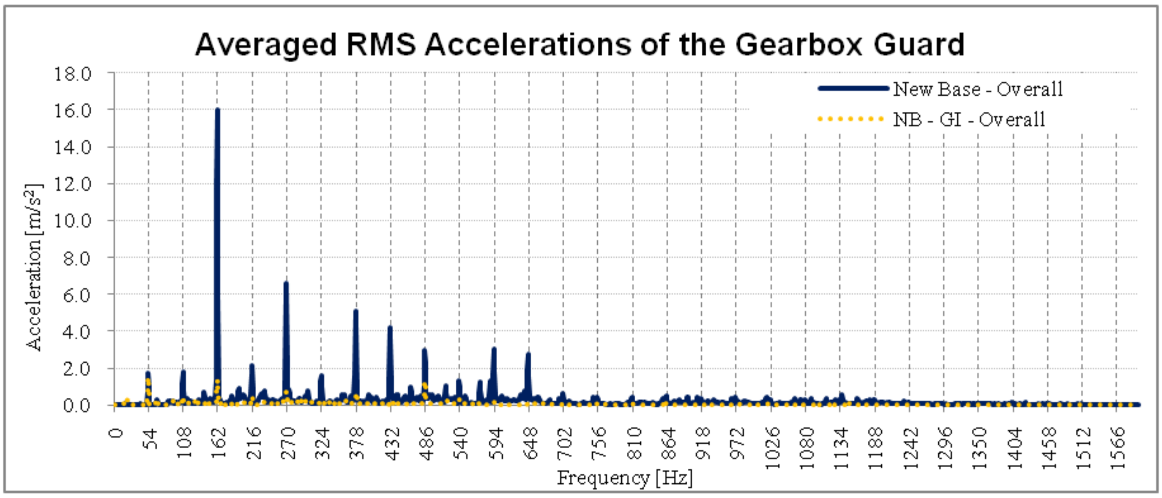


Figure 76 Accelerations of the gearbox guard – overall

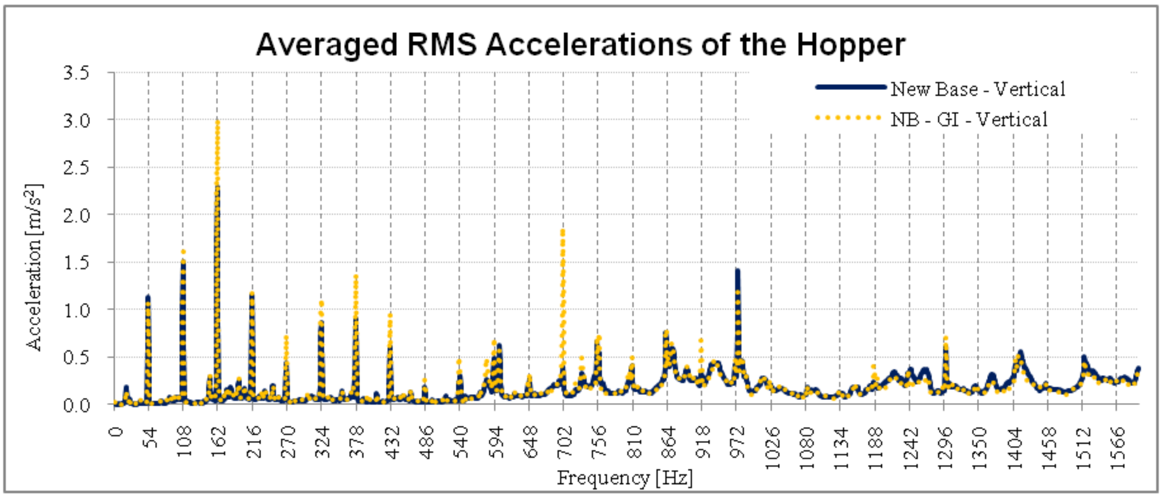


Figure 77 Accelerations of the hopper – vertical

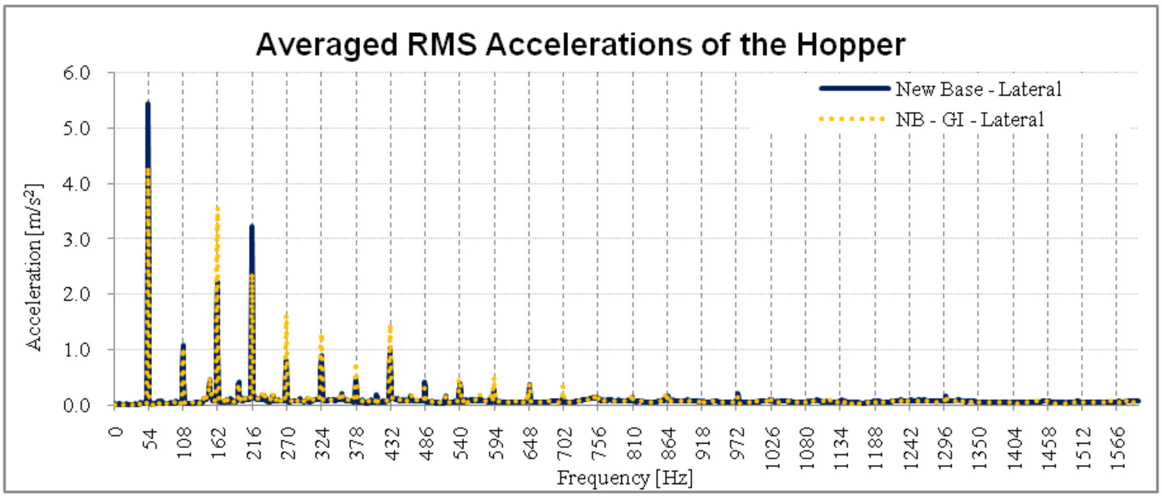


Figure 78 Accelerations of the hopper – lateral

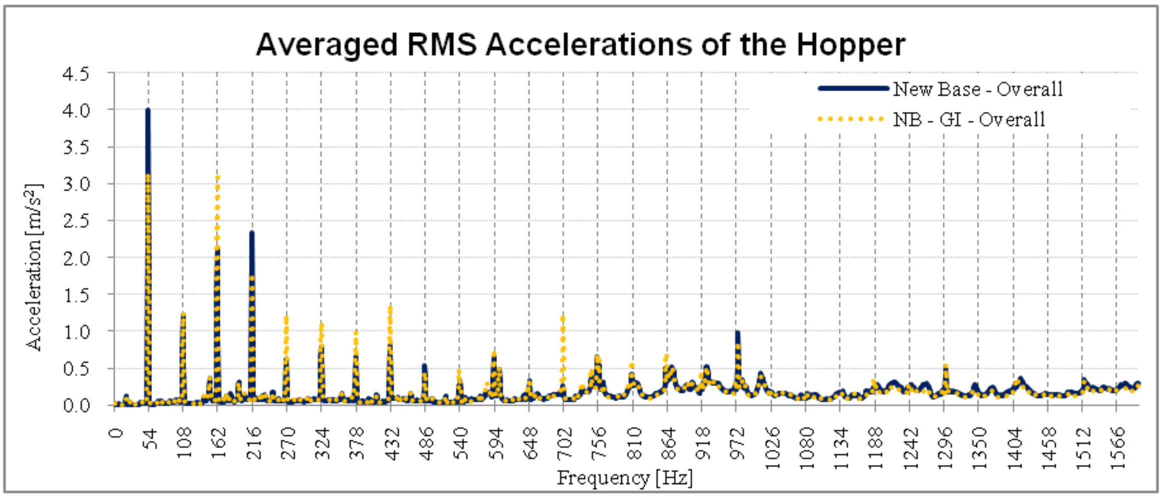


Figure 79 Accelerations of the hopper – overall

6.3 Discussion

The most significant change in the sound power produced by the decanter was the 12 dB reduction in the 160 Hz one-third octave band. This was directly attributable to the isolation of the gearbox guard. The intensity scans showed that the regions associated with the gearbox guard had significant, greater than 17 dB average, sound intensity reductions. The acceleration measurements on the gearbox guard also showed a significant reduction at 162 Hz. The overall sound power reduction of 2.6 dB showed that the gearbox guard was contributing 45% of the overall sound power.

As the 160 Hz one-third octave band was the most significant contributor to the overall sound power it must be reduced in order to lower the overall sound power level. In the previous investigation it was determined that the gearbox guard was the major contributor of sound power in the 160 Hz one-third octave band. Isolating the gearbox guard from the base has resulted in the 160 Hz one-third octave band no longer being a major contributor to the sound power of the decanter.

The next two highest levels in one-third octave bands, 100 and 200 Hz, have also decreased by 1.3 and 0.8 dB respectively. This is due to significant reductions in accelerations in the gearbox guard and no increase in vibrations in the hopper for the harmonic frequencies of 108 and 216 Hz. As the 100 and 200 Hz one-third octave bands are now the two most significant contributors to the overall sound power, their small decrease is significant in achieving the overall 2.6 dB sound power reduction.

When the gearbox guard was isolated from the original base there was a 0.6 dB increase in overall sound power from the decanter. Isolating the gearbox guard from the new base has resulted in a 2.6 dB reduction in sound power. This indicates that the new base is more effective in absorbing the additional vibrational energy, which was going to the gearbox guard, without generating as much sound power. Isolating the hopper from the base should also generate additional loses for the same reasons that isolating the gearbox guard did.

6.4 Conclusion

Isolation of the gearbox guard was successful in reducing the overall sound power by 2.6 dB. The 160 Hz one-third octave band was the most significant contributor to the overall sound power but this was reduced by 12 dB through isolation of the gearbox guard.

The 160 Hz one-third octave band is no longer a significant contributor to the overall sound power of the decanter.

The new base has proved more effective than the original base in absorbing an increase in vibrational energy without an increase in sound power. Isolating the hopper from the base should also result in reductions in sound power from the decanter and should be the next area of investigation.

7 Modification – Hopper Isolation

The hopper was isolated from the new polymer concrete base by the use of isolation pads, as illustrated in Figure 80. The positioning of the hopper remained unchanged. The gearbox guard remained isolated from the base, see Section 6. This configuration will be referred to as NB-HI.

a b

Figure 80 (a-b) Photographs of the isolation of the hopper from the new base

7.1 Results

The overall sound power of the decanter calculated from two sound intensity scans over the decanter is shown in Figure 81. The scans were undertaken by the method described in Section 2.7. The two scans produced very consistent results, less than 0.4 dB variation, except for the low frequency bands and 200 to 315 Hz one-third octave bands which had about a 1 dB difference. Therefore the scans were deemed to be a fair representation of the sound power that the decanter produces. The overall sound power of the decanter was 100.7 dB.

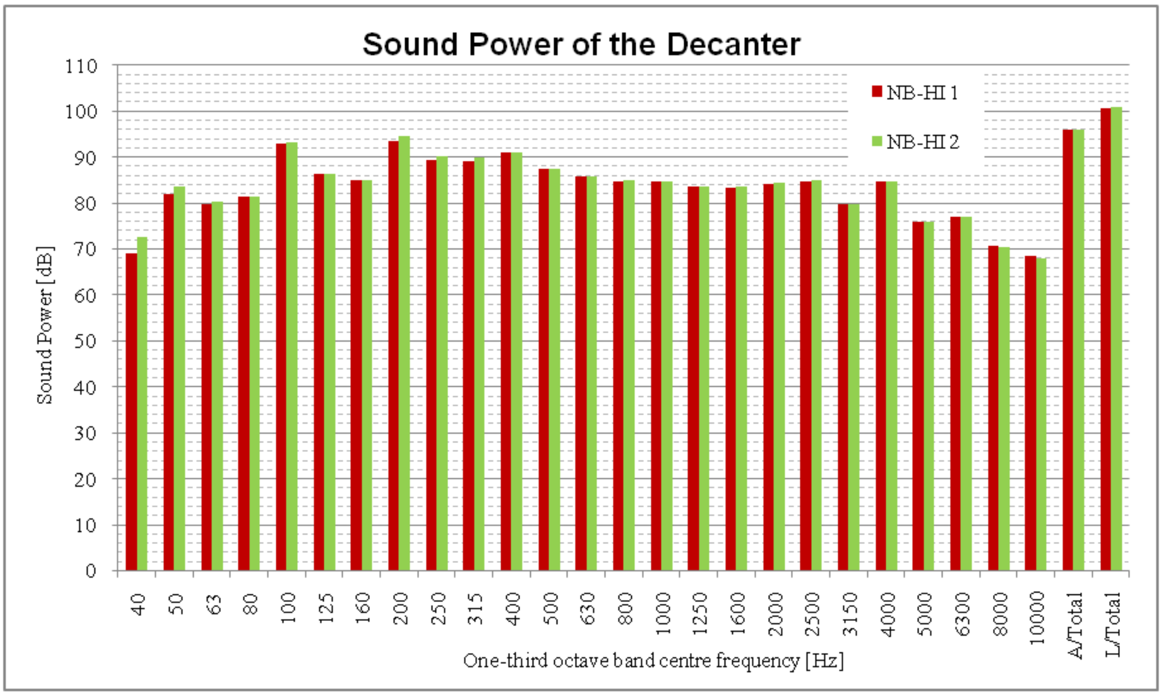


Figure 81 Sound power of the decanter

Vibration acceleration measurements were taken after the two scans were completed. The overall results for the acceleration measurements are shown in Table 18 and the spectrum in Figure 82. The original configuration measurements are described in Section 3.1.3 and the previous configuration, NB-GI, measurements are described in Section 6. The acceleration measured on the hopper had reduced by about 60%. The accelerations for all the driven harmonic frequencies had reduced except for the 108 Hz harmonic which increased. The overall accelerations measured within the gearbox guard decreased slightly and the base had a slight overall increase in measured accelerations.

Table 18 Overall acceleration results

	RMS Average Accelerations [m/s^2]			
Configuration	Base	Hopper	GB Guard	Total
Original	0.15	0.46	1.26	0.70
NB - GI	0.12	0.25	0.12	0.18
NB - HI	0.14	0.15	0.08	0.13

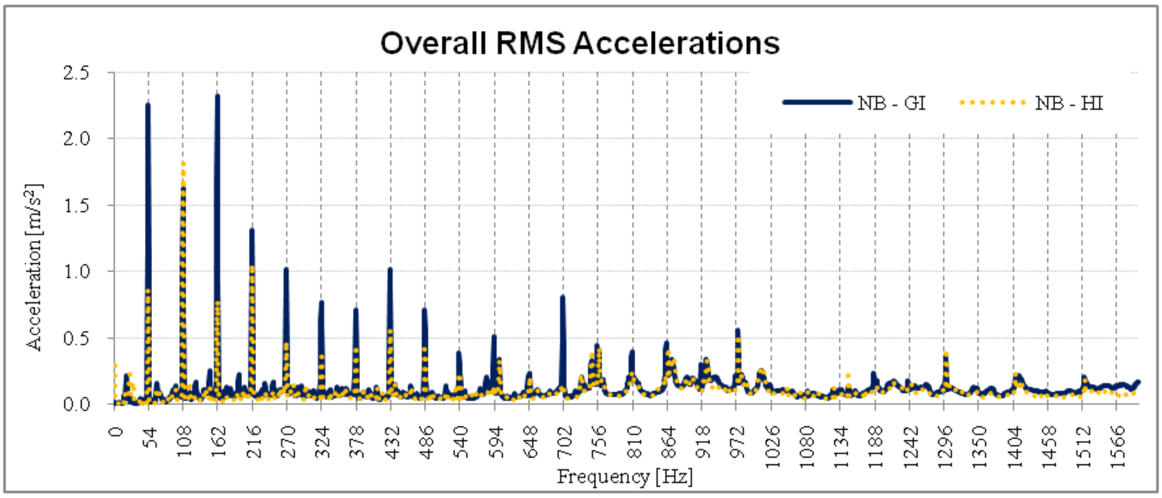


Figure 82 Total averaged RMS accelerations of driven harmonic frequencies

7.2 Comparisons

7.2.1 Sound Intensity Measurements

The comparison of the intensity scans of the current decanter configuration, NB-HI, and the previous configuration, NB-GI, are shown in Figure 83 and Tables 19 and 20. Figure 83 shows that the sound power of the decanter had decreased across all one-third octave bands. The largest reductions were in the 50 and 160 Hz one-third octave bands. These bands contain the first and third harmonic of the driven bowl frequency. Isolating the hopper resulted in a 2.7 dB reduction in the sound power emitted by the decanter. This indicated that the hopper was the source of nearly half the sound power of the decanter.

The reduction in sound intensity from the decanter was from all regions, as shown in Table 19. The 200 Hz one-third octave band showed the typical result for most of the one-third octave bands and was the band with the highest sound power, see Table 20. The sound intensity reductions were not restricted to regions directly related to the hopper, with all sides showing similar reductions.

A comparison between the current configuration and the original configuration is shown in Figure 84. The changes from the original configuration were a polymer concrete base and isolation of both the gearbox guard and hopper. There were significant reductions in the sound power in the one-third octave bands that relate to the first three driven harmonics. There were only minor changes in sound power above 400 Hz.

A comparison between the current configuration and the new base configuration is shown in Figure 85. The changes from the new base configuration were the isolation of both the gearbox guard and hopper. The most significant reduction was in the sound power in the 160 Hz one-third octave band with a reduction of 17.4 dB. There were only minor changes in sound power at 315 Hz and above.

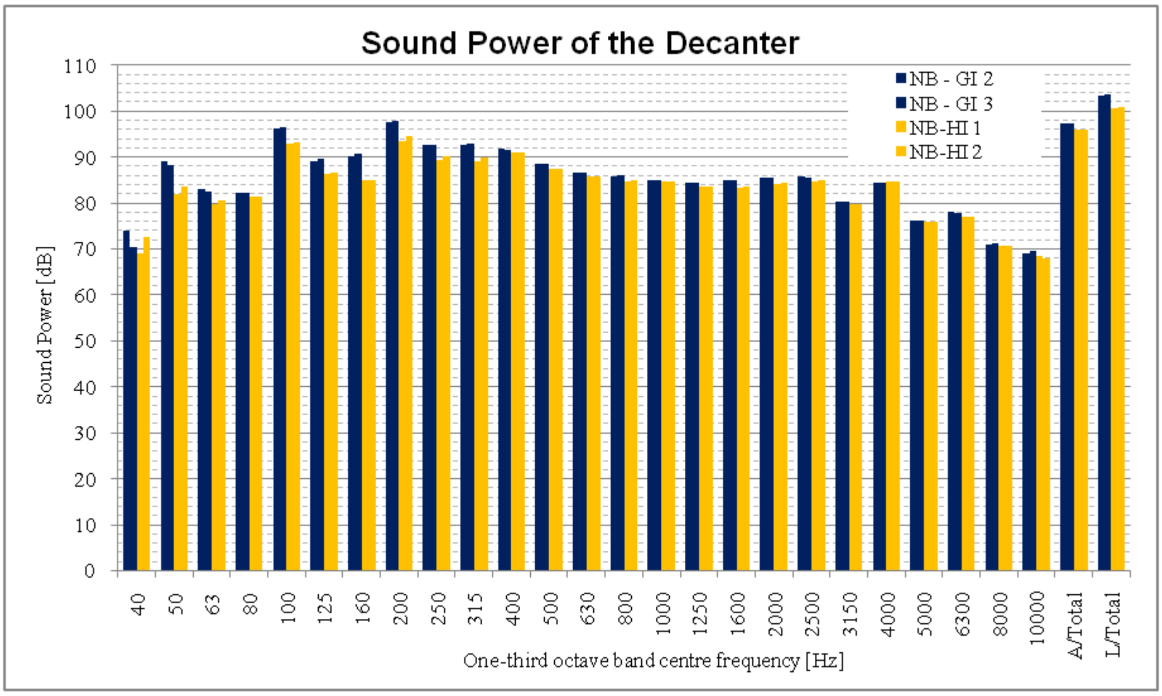


Figure 83 Sound power comparison

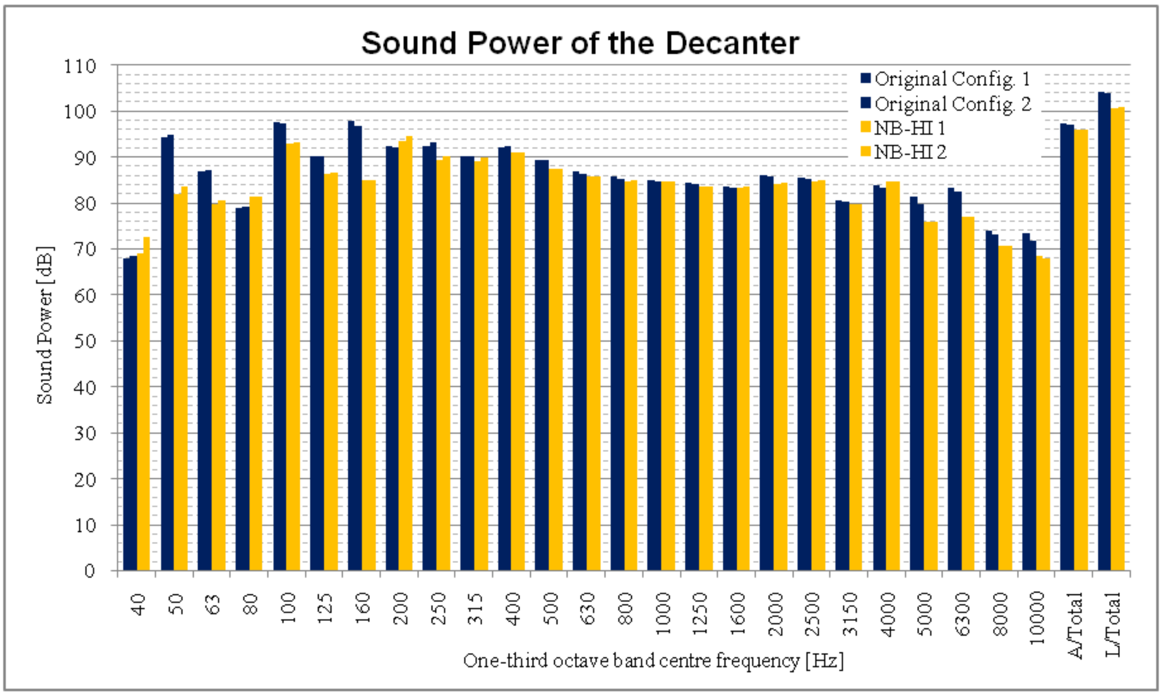


Figure 84 Sound power comparison

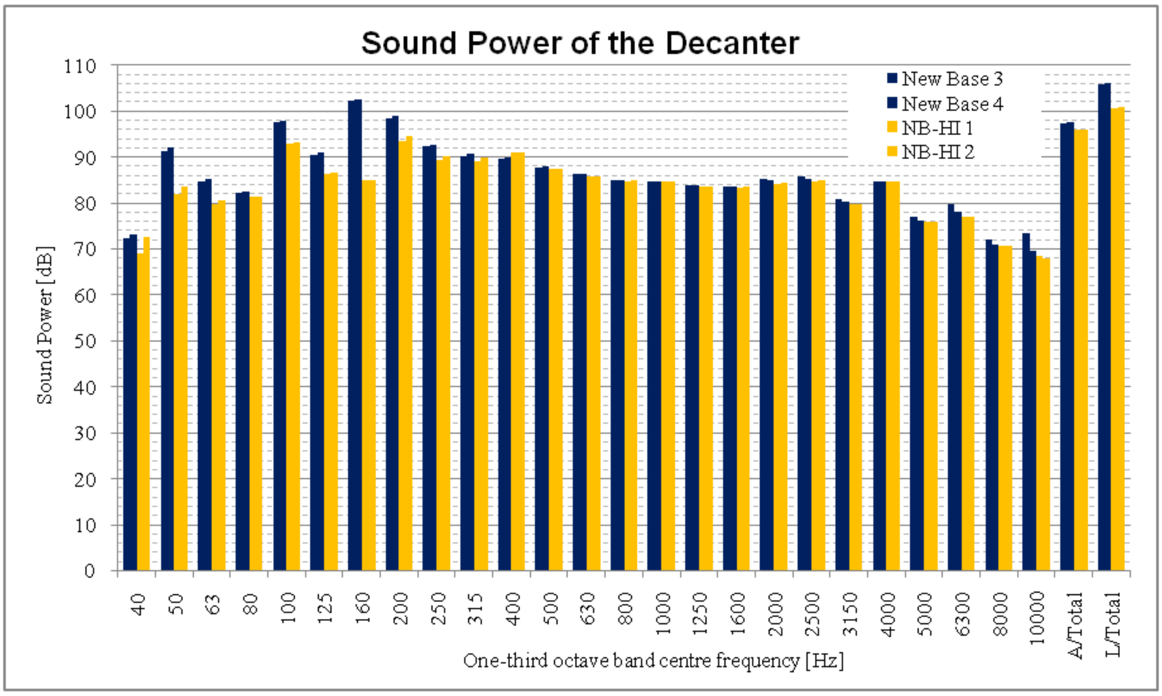


Figure 85 Sound power comparison

Table 19 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-GI2	88.2	90.3	92.0	85.6		93.9	95.4	94.0	86.6	92.2
NB-GI3	87.5	90.0	93.2	87.2		93.8	94.8	94.3	87.8	92.4
NB-HI1	84.0	86.7	87.1	80.1		90.0	92.8	92.0	85.4	89.3
NB-HI2	85.5	88.0	87.7	79.0		91.5	93.2	92.5	85.8	90.0
Ave Diff.	-3.1	-2.8	-5.2	-6.9		-3.1	-2.1	-1.9	-1.6	-2.7
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-GI2	85.8	91.2	90.3	88.6		85.7	95.7	94.2	89.1	91.9
NB-GI3	86.3	91.4	90.6	88.6		83.1	96.0	94.7	89.5	92.2
NB-HI1	82.9	87.8	87.6	85.3		84.2	92.1	91.7	87.0	88.9
NB-HI2	83.1	88.3	87.9	85.6		84.5	92.6	91.7	87.2	89.2
Ave Diff.	-3.1	-3.3	-2.7	-3.2		-0.1	-3.5	-2.8	-2.2	-3.0
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-GI2	88.2	89.3	89.6	91.3	90.4			87.0	88.2	87.7
NB-GI3	87.5	88.8	89.5	91.7	90.6			87.1	89.3	88.4
NB-HI1	86.3	88.1	87.4	89.4	88.6			83.2	86.5	85.3
NB-HI2	86.2	88.3	88.4	90.4	89.4			82.7	86.8	85.4
Ave Diff.	-1.6	-0.9	-1.6	-1.6	-1.5			-4.1	-2.1	-2.7
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-GI2	90.0	90.7	89.9	86.9	89.6					91.2
NB-GI3	89.8	90.7	89.9	87.0	89.6					91.4
NB-HI1	85.5	86.9	86.8	85.7	86.4					88.4
NB-HI2	86.5	87.3	87.0	85.6	86.7					88.9
Ave Diff.	-3.9	-3.6	-3.0	-1.3	-3.1					-2.7

Table 20 Comparison of sound intensity measurements for the 200 Hz one-third octave band [dB]

One-third octave band centre frequency:				200	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-GI2	83.1	82.3	82.5	0.0		91.4	91.0	85.9	77.3	86.5
NB-GI3	81.2	81.1	82.8	0.0		91.9	91.3	86.5	79.5	86.7
NB-HI1	75.5	76.6	77.5	0.0		86.7	86.9	85.0	78.1	82.8
NB-HI2	79.5	80.7	79.6	0.0		88.1	87.3	85.1	79.9	83.7
Ave Diff.	⇩ -4.7	⇩ -3.0	⇩ -4.1	-		⇩ -4.3	⇩ -4.1	⇩ -1.2	⇨ 0.6	⇩ -3.4
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-GI2	73.6	85.5	85.1	81.7		0.0	90.1	87.7	77.8	85.6
NB-GI3	69.3	86.1	86.1	81.6		0.0	90.3	88.8	81.1	86.1
NB-HI1	70.5	80.9	80.9	77.6		0.0	86.0	83.7	80.1	81.6
NB-HI2	69.3	81.7	81.7	78.3		0.0	87.2	84.6	80.0	82.6
Ave Diff.	⇩ -1.5	⇩ -4.5	⇩ -4.3	⇩ -3.7		-	⇩ -3.6	⇩ -4.1	⇨ 0.6	⇩ -3.8
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-GI2	82.0	83.0	85.7	88.5	86.9			81.6	77.9	80.0
NB-GI3	80.4	77.4	85.6	88.8	86.8			79.7	79.8	79.8
NB-HI1	78.0	79.0	82.2	84.5	82.9			76.0	77.7	77.0
NB-HI2	77.1	78.4	84.8	86.1	84.4			74.5	79.1	77.5
Ave Diff.	⇩ -3.7	⇩ -1.5	⇩ -2.2	⇩ -3.4	⇩ -3.2			⇩ -5.4	⇨ -0.4	⇩ -2.6
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-GI2	87.3	85.8	79.6	81.0	84.1					85.5
NB-GI3	87.0	85.7	80.9	81.4	84.1					85.7
NB-HI1	80.2	77.3	77.7	77.4	78.0					81.4
NB-HI2	82.6	79.9	78.5	78.1	79.8					82.5
Ave Diff.	⇩ -5.8	⇩ -7.2	⇩ -2.2	⇩ -3.5	⇩ -5.2					⇩ -3.7

7.2.2 Acceleration Measurements

The Figures 86 to 94 show the vibration acceleration measurements of the three main components under investigation on the decanter. The acceleration measurements have been combined by taking the ‘root of the mean of the squares’ (RMS) of the various individual measurements.

The accelerations measured on the base generally increased, particularly at 108 Hz. The hopper shows a significant decrease in the accelerations measured for frequencies up to 702 Hz with an overall reduction of 40 %. The accelerations of the gearbox guard also decreased, particularly the lateral accelerations, except for several vertical measurements in the mid-frequencies, 432 - 540 Hz.

At 108 Hz, the second harmonic, the measurements were twice as large as at the next highest harmonic which is the forth, 216 Hz. The vibrational acceleration measurements at 108 Hz were equal for the lateral and vertical directions. The 216 Hz vibrational acceleration measurements were highest in the vertical direction.

The lateral acceleration of the hopper had significantly diminished. The vertical accelerations also significantly diminished except for the measurements in the top half of the spectrum, 703 to 1512 Hz. It was not likely that the vibrations for these frequencies come through the isolation pads given the reductions at the lower frequencies.

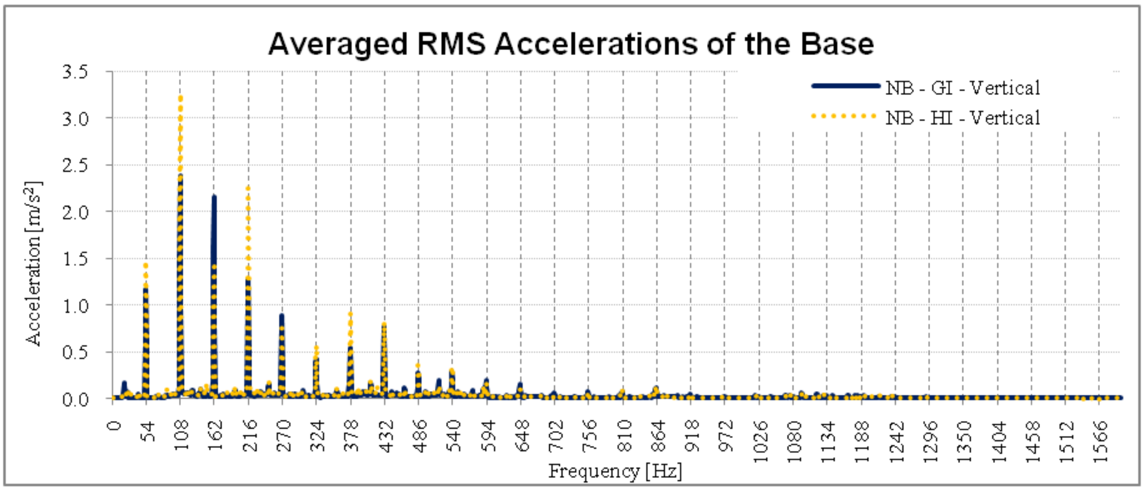


Figure 86 Accelerations of the base – vertical

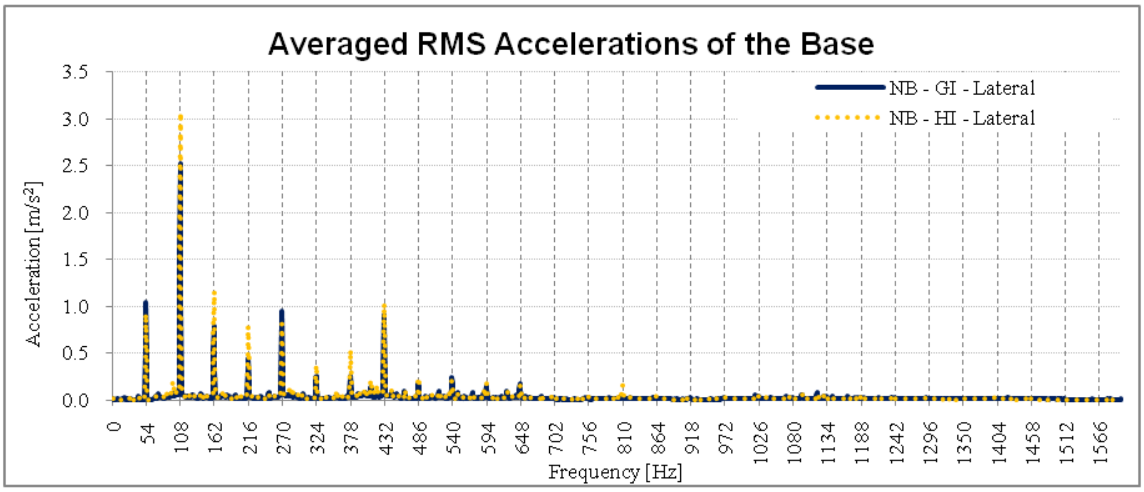


Figure 87 Accelerations of the base – lateral

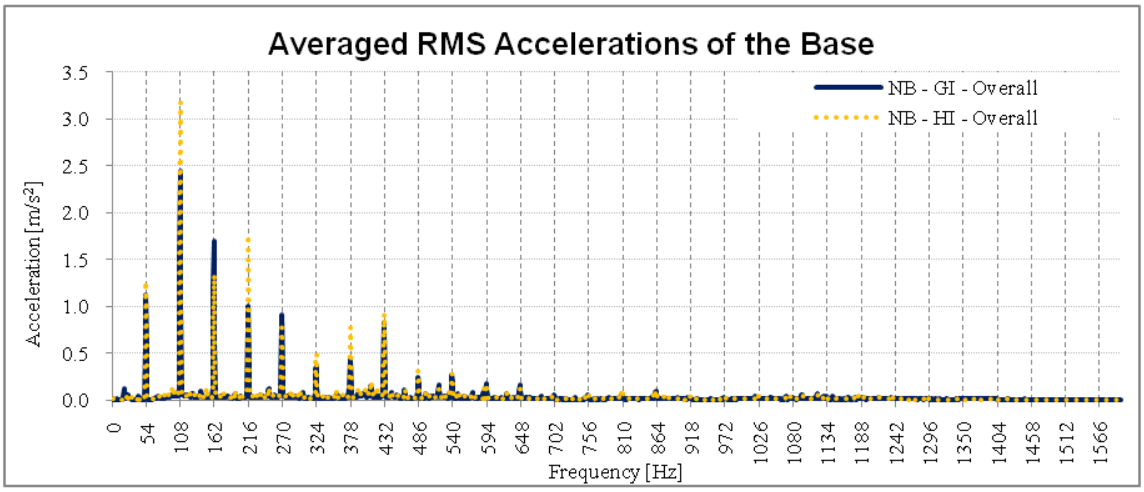


Figure 88 Accelerations of the base – overall

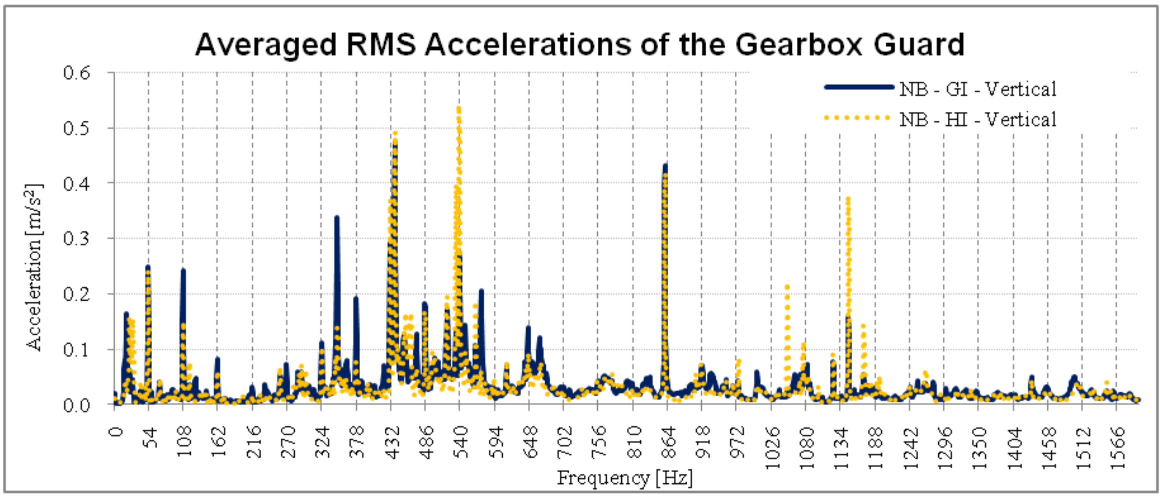


Figure 89 Accelerations of the gearbox guard – vertical

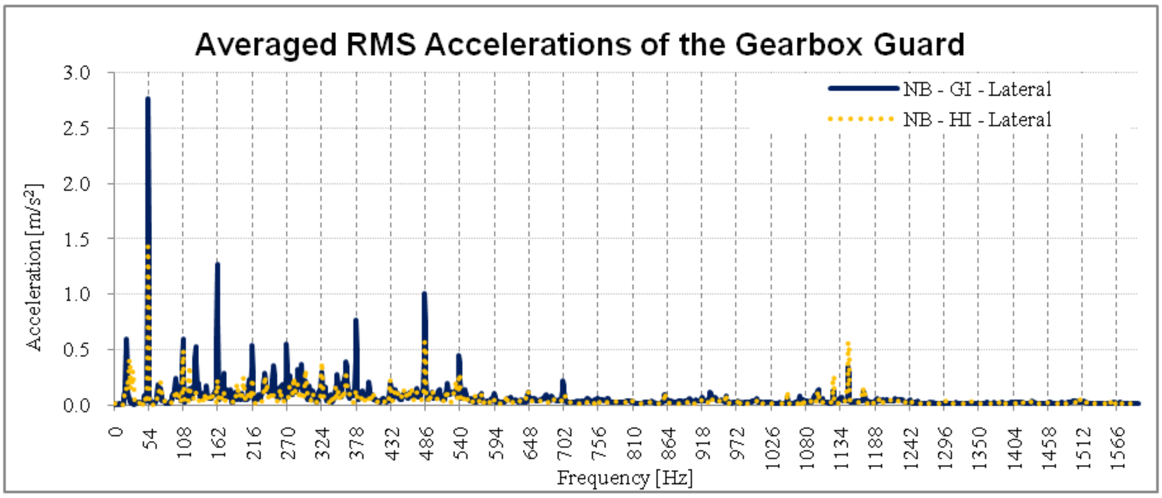


Figure 90 Accelerations of the gearbox guard – lateral

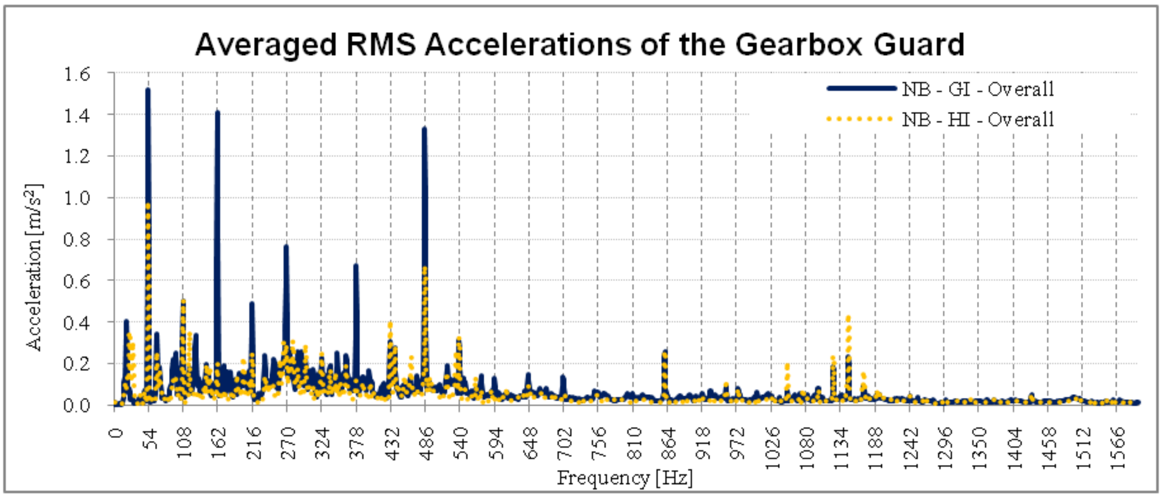


Figure 91 Accelerations of the gearbox guard – overall

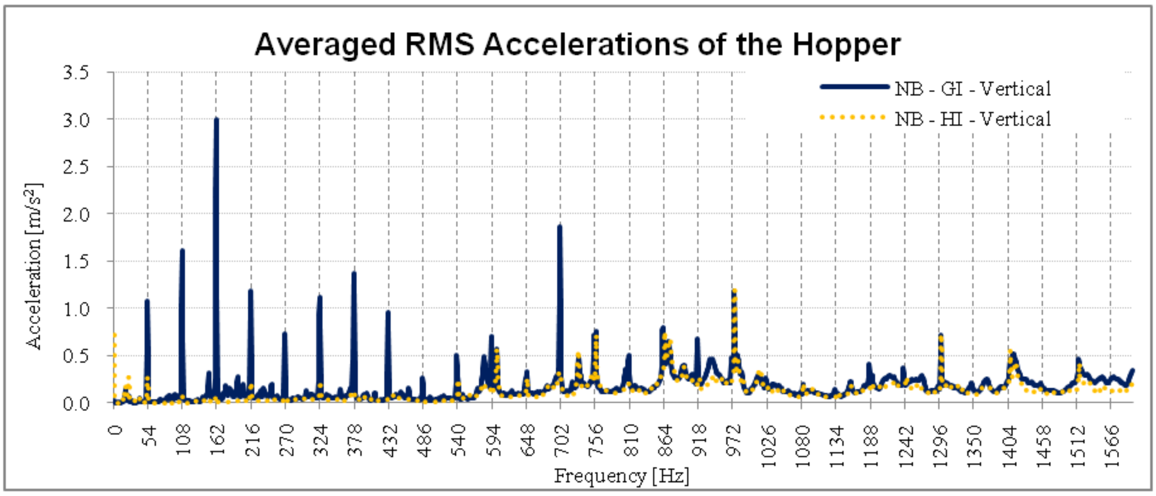


Figure 92 Accelerations of the hopper – vertical

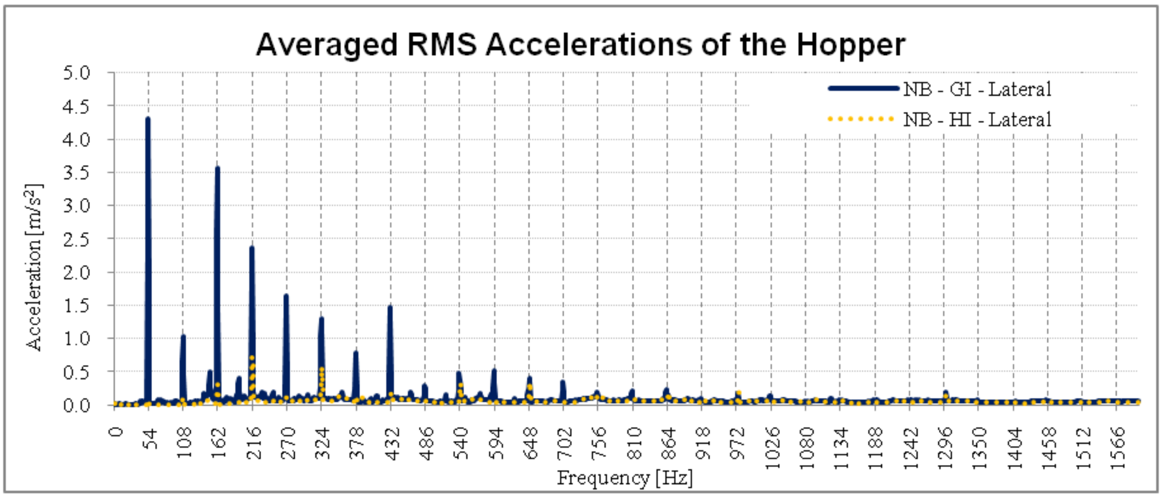


Figure 93 Accelerations of the hopper – lateral

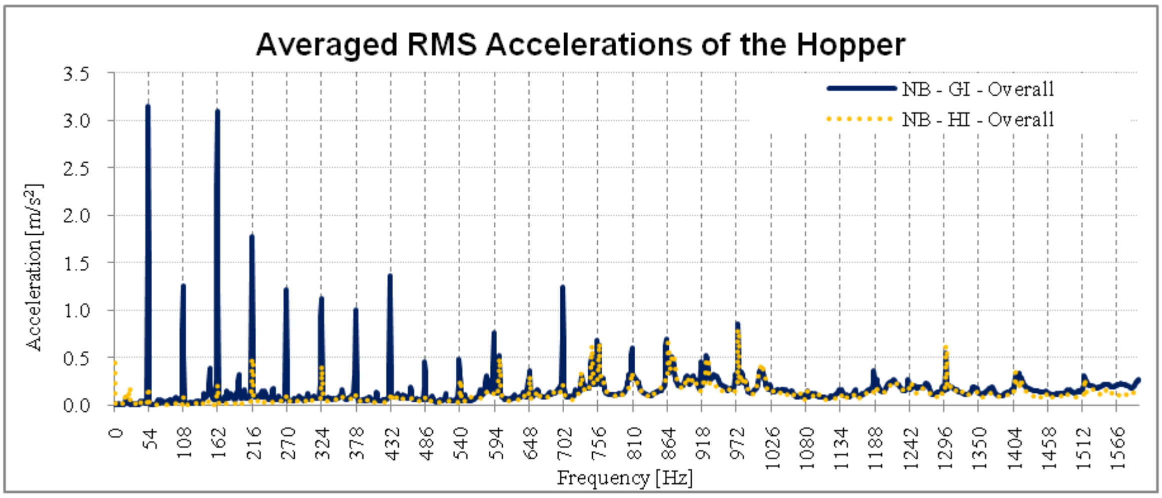


Figure 94 Accelerations of the hopper – overall

The comparison between the current configuration and the original configuration of overall results for acceleration measurements are shown in Figures 95 to 97. It shows that the vibration levels for the base, gearbox guard and hopper had been reduced significantly. The overall vibrational levels had been reduced from 0.70 to 0.13 m/s^2, 80 % reduction. The only increases in measured vibrational levels were for the base and these were only for the frequencies 108 Hz and 216 Hz.

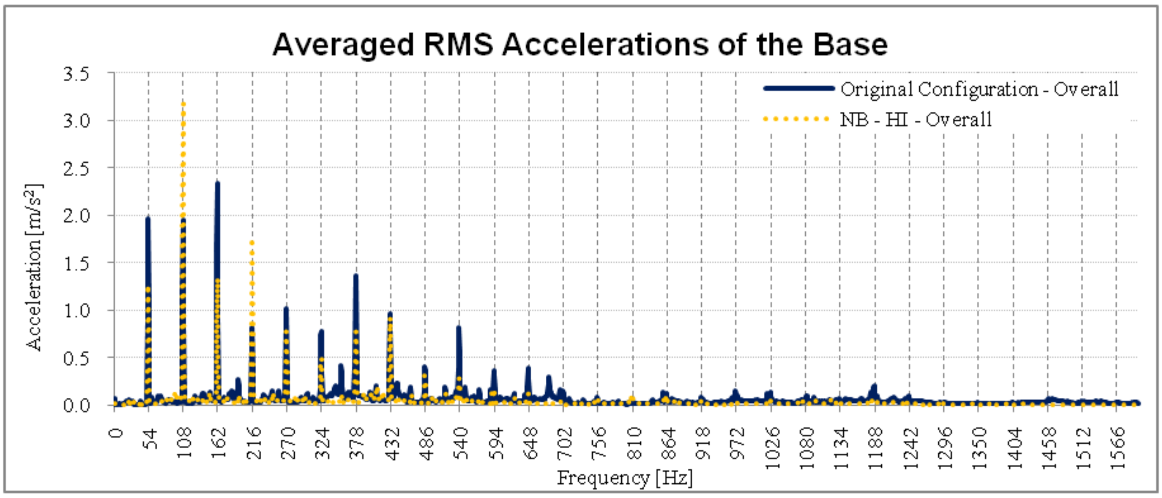


Figure 95 Accelerations of the base – overall

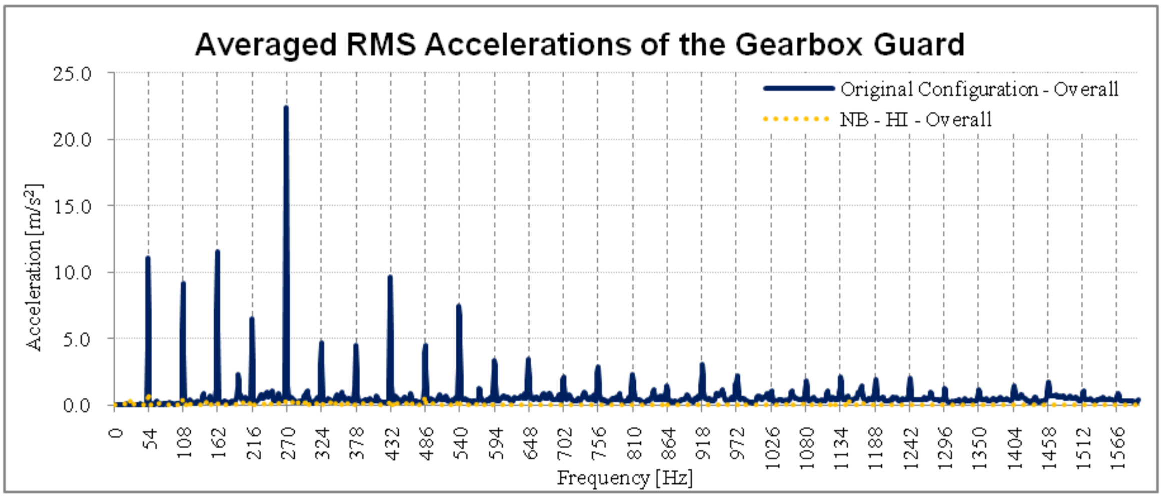


Figure 96 Accelerations of the gearbox guard – overall

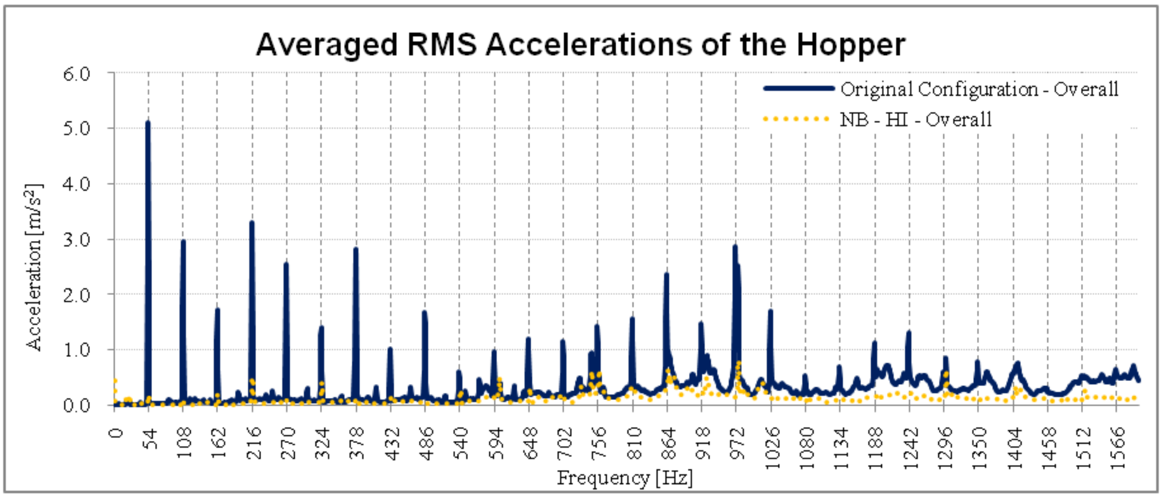


Figure 97 Accelerations of the hopper – overall

7.3 Discussion

Isolating the hopper produced significant reductions, 2.7 to 6.0 dB, in the 50 Hz to 315 Hz one-third octave bands, excluding the 80 Hz band. Most of the remaining one-third octave bands had sound power reductions between 0.5 and 1.5 dB. The result was an overall reduction of 2.7 dB with the A-weighted total reducing 1.3dB.

The regions with the highest sound intensity were Front 6 and 7 and Back 6 and 7.These regions are the lower middle front and back surfaces. These regions are not directly associated to a component but it was likely that the noise source was coming from the hopper, base or back-drive motor. The 100 Hz and 200 Hz one-third octave bands had the highest sound power levels. Front 6 and 7 and Back 6 and 7 are also the highest regions of sound intensity in the 100 Hz and 200 Hz one-third octave bands.

The only vibrational harmonic frequency in the 100 Hz one-third octave band was 108 Hz. For 108 Hz, the vibration acceleration levels had increased within the base to 3.2 m/s^2, remained the same in the gearbox guard (0.5 m/s^2) and essentially disappeared from the hopper (0.1 m/s^2). This indicates that the base was the likely source for vibration induced sound for this one-third octave band.

For the 200 Hz one-third octave band, 216 Hz was the only vibrational harmonic within the band. The vibrational levels, for 216 Hz, had increased within the base to 2.2 m/s^2. Both the gearbox guard and hopper showed reductions to 0.2 m/s^2 and 0.5 m/s^2

respectively. This indicates that the base was also the likely source for vibration induced sound for this one-third octave band.

The vibrations in the base at 108 Hz and 216 Hz were the only measured accelerations over 2 m/s^2. As the corresponding 100 Hz and 200 Hz one-third octave bands had the highest sound power levels, the vibration of the base remained a likely contributor to the overall sound power of the decanter. In order to reduce decanter vibration induced sound power, the vibration of the base at 108 Hz and 216 Hz needs to be reduced.

Except for the spikes at 702 Hz and 918 Hz where there was a reduction, the measured vertical vibrational accelerations for the hopper were unchanged for frequencies above 594 Hz. The source of energy for these frequencies was unlikely to have come through the isolation pads, given that there had been significant reductions in the vibration levels for the lower frequencies due to the introduction of the isolation pads. The only other likely source of the vibration was air turbulence due to the rotating bowl assembly. The bowl assembly would induce air movement due to the pumping action of the rotating auger and due to the outer surface of the bowl not being smooth.

The averaged vibrational levels were very low (less than 0.5 m/s^2) in the base above 432 Hz and in the gearbox guard above 540 Hz (excluding the small spike at 1147 Hz). The only vibrational levels above 0.5 m/s^2 for the hopper were likely due to air turbulence from the spinning bowl. This indicated that decanter vibration was not the primary source of noise for these higher frequencies.

There was no reduction in sound power above the 400 Hz one-third octave band with the new base configuration. The measured vibration acceleration in the base had not increased above 432 Hz. This indicates that the gearbox guard and hopper were not significant contributors to the decanter sound power above the 400 Hz one-third octave band with the new polymer concrete base.

The noise level comparison between the original configuration and the current configuration showed it to be essentially unchanged for the 315 Hz to 1600 Hz one-third octave bands. The vibrational levels between these two configurations showed that there had been significant reductions in measured vibrational levels for all frequencies, except at 108 Hz and 216 Hz on the base. This indicated that decanter vibrations were not the main source of noise in the 315 Hz to 1600 Hz one-third octave bands.

7.4 Conclusion

Isolation of the hopper resulted in a reduction of 2.7 dB and 1.3 dB(A) in sound power. This indicates that the hopper was the source of nearly half the sound power of the decanter. The 50 Hz to 315 Hz one-third octave bands, excluding the 80 Hz band, showed reductions between 2.7 and 6.0 dB. There were only minor reductions in sound power for the 400 Hz one-third octave band and above. The gearbox guard and hopper were not significant sources of sound power for noise over 400 Hz one-third octave band with the new base decanter configuration.

The two highest measured vibration accelerations were at 108 Hz and 216 Hz, in the base. These frequencies are in the 100 Hz and 200 Hz one-third octave bands which contain the two highest contributions to the total decanter sound power. The source of the highest sound intensity was from the regions of the lower middle of the front and back surfaces.

Vibrations within the hopper were reduced by 40 %. The vibrations within the hopper were likely due to air turbulence from the rotating bowl. Decanter vibrations were not the main source of noise in the 315 Hz to 1600 Hz one-third octave bands.

Further reduction in the decanter sound power requires that:

- The vibrations of 108 Hz and 216 Hz within the base are reduced.
- The air turbulence from the rotating bowl is reduced.
- The source of the sound intensity for the lower middle regions of the front and back are be determined.

8 Modification – Tuned Mass Dampers

The tuned mass damper (TMD) decanter test configuration was based on that used for assessment of the hopper isolation configuration see Section 7. The gearbox guard and hopper remained isolated from the base and TMD's were added. This configuration will be referred to as NB-TMD. The original design of the TMD's was to mount two cantilevered beams, with 5 kg weights at their ends, onto a central mount. One end was to be tuned to 108 Hz and the other end tuned to 216 Hz. As the base weighed around 400 kg, the (8 x) 5kg weights at the end of the TMD's was considered the minimum feasible weight in order to draw enough energy out of the base to be effective. TMD's are normally at least 10 % of the weight of the structure that they are damping [7].

The natural frequency of a single degree of freedom system is defined by $f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$

Where : f_n = natural frequency [Hz]

k = spring stiffness [N/m]

m = mass of the system [kg]

For a natural frequency of 108 Hz or 216 Hz and mass of 5 kg, the spring stiffness would need to be 2.3 or 9.3 MN/m respectively. In order to achieve this stiffness, a steel bar was selected as the spring. Figure 98 shows a cantilevered bar with a mass on the end. This was the configuration selected to be used on the decanter. The equation [16] for the natural frequency of this system is $f_n = \frac{1}{2\pi}\sqrt{\frac{3EI}{l^3(0.23m_1+m_2)}}$

Where : f_n = natural frequency [Hz]

E = Youngs Modulus [200 GPa]

I = Second moment of area of the bar [m^4]

l = length of the canter lever [m]

m_1 = mass of the bar [kg]

m_2 = mass of the end weight [5 kg]

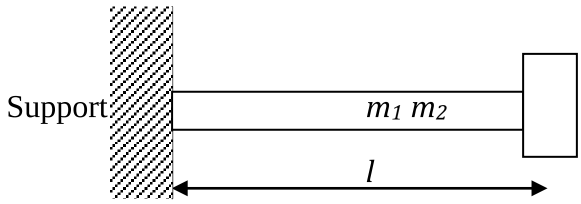


Figure 98 Tuned mass damper

Four standard bar diameters were then analysed and the results are shown in Figure 99. Bars with diameters 24, 32, and 50 mm are able to be tuned to both 108 Hz and 216 Hz with bar lengths between 100 mm and 450 mm. The bar of diameter 50 mm was chosen due to the longer bar length and the resulting ability to fine tune the TMD's

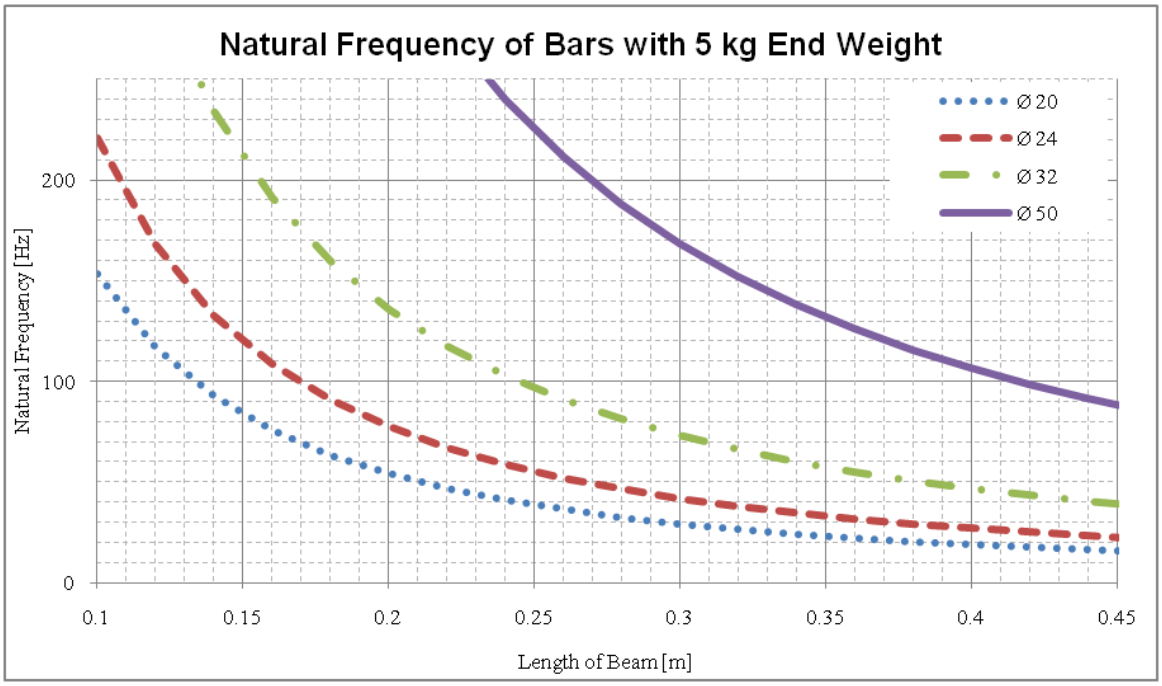


Figure 99 Natural frequencies of bars with a 5 kg end weight

The TMD's were then designed to be attached to the decanter base at its four corners where there were pre-existing M16 bolt holes. The design and assembly of the TMD's are shown in Appendix A. The design was then modelled in ANSYS to verify the calculations and the results are shown in Figure 100. The support had the top and bottom surfaces fixed in the ANSYS model to simulate the clamping effect onto the base. The natural frequencies determined using ANSYS agreed well with the calculated results.

Four sets of tuned mass dampers (TMD) were manufactured and attached to the decanter. There were several attempts to tune them to 108 Hz and 216 Hz as designed. This was not possible as the support was not stiff enough to ensure that each end of the assembly acted independently. It was then decided to alter the design of the assembly. As the support was not rigid, the first configuration was symmetrical and tuned to 54 Hz, in the vertical direction. This resulted in the second natural frequency in the lateral direction being just above 108 Hz. The second configuration was tuned to 108 Hz, in the vertical direction, with the mount stiffened by using a wedge under the far end of the bar.

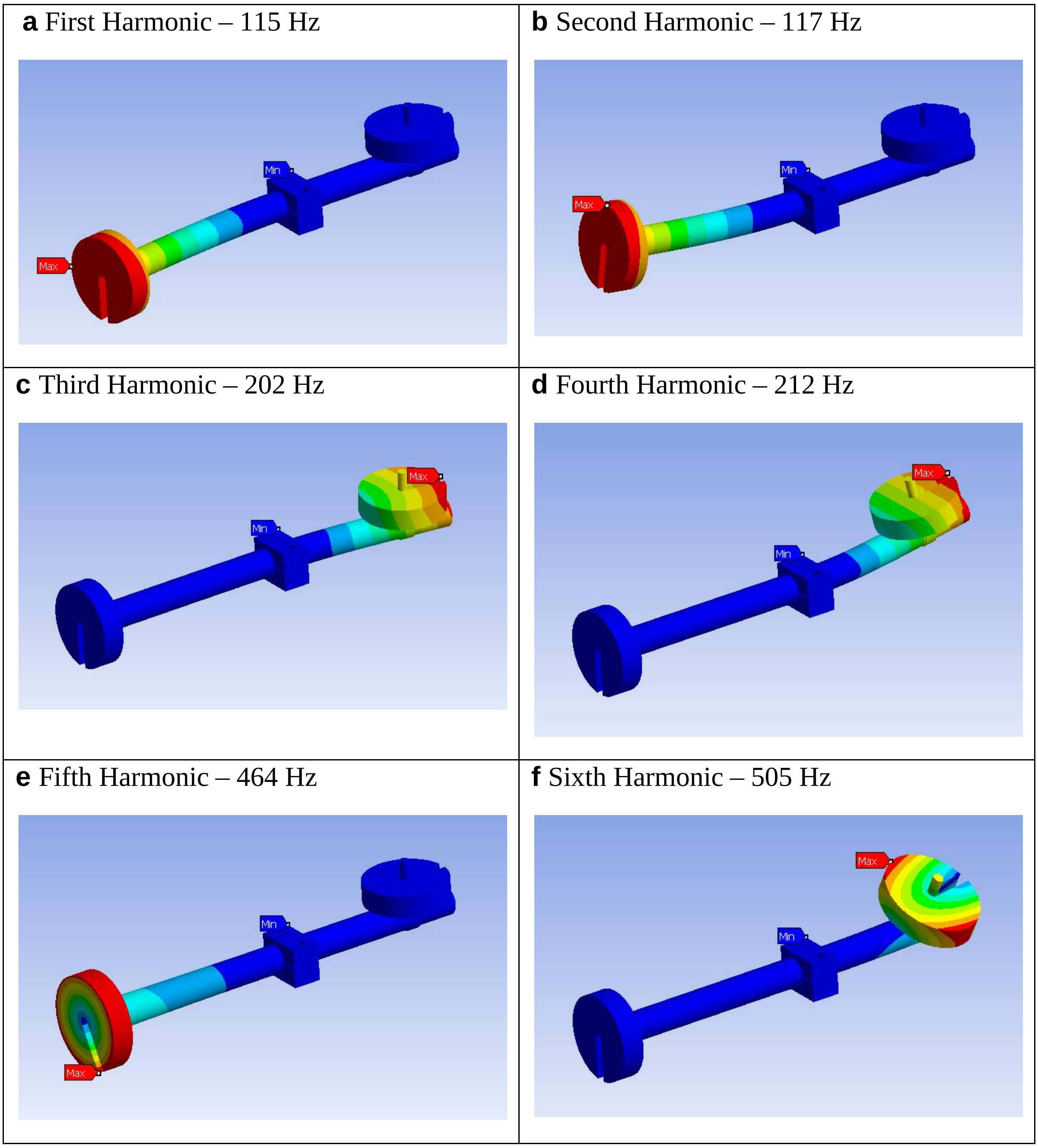


Figure 100 (a – f) ANSYS analysis of the tuned mass dampers

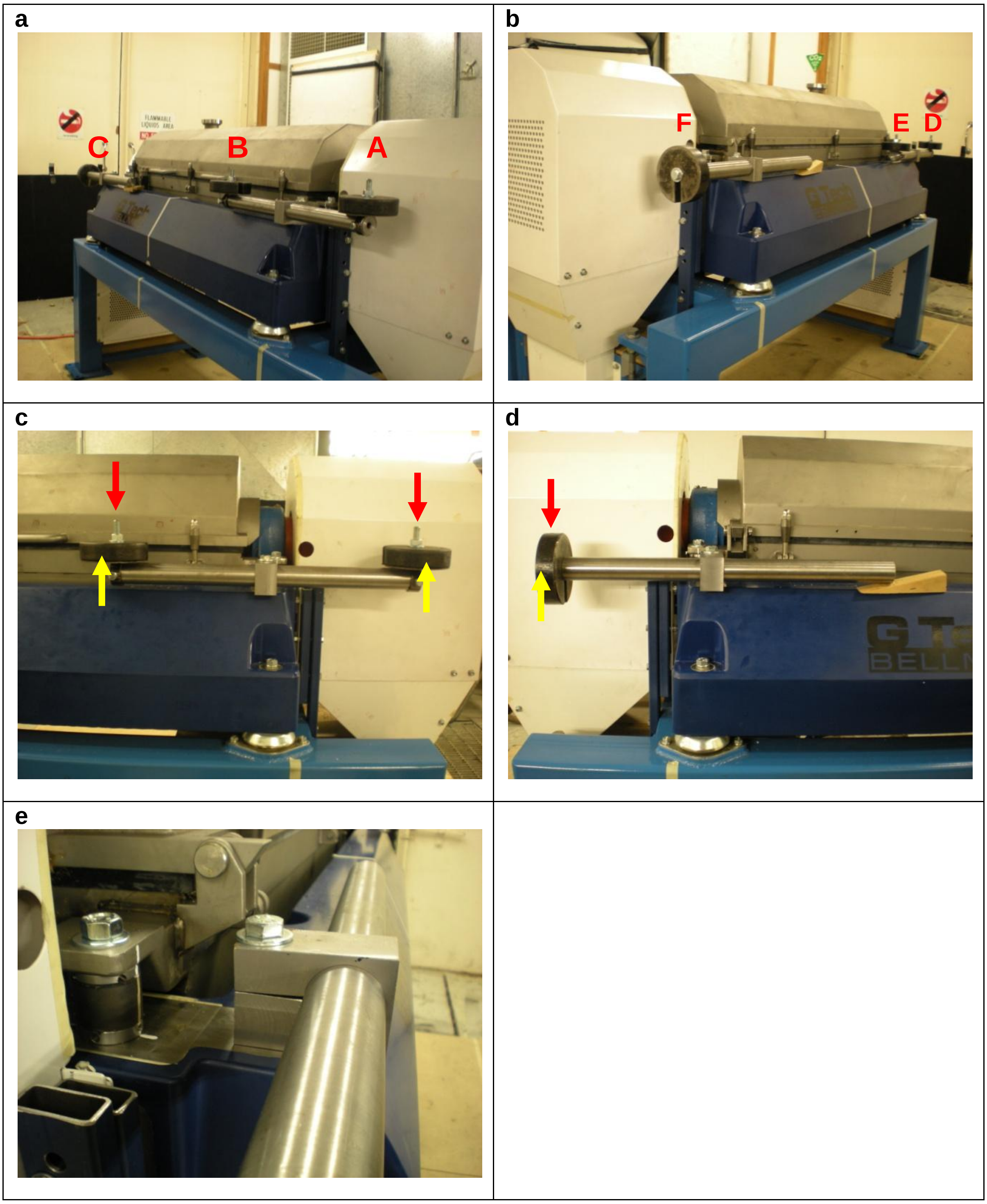


Figure 101 (a-e) Photographs of the tuned mass dampers

The TMD's were attached to the base as shown in Figure 101. The positioning and labelling of the TMD's are shown in Figures 101 (a) and (b). Figure 101 (c) shows the first configuration, a symmetrical assembly with the support in the middle and 5 kg weights equally spaced from the support. Figure 101 (d) shows the second configuration which has a 5 kg weight at one end of the bar and a wedge at the other end of the bar to stiffen the mounting of the TMD. The red arrows in Figures 101 (c) and (d) show the location for the vertical acceleration measurements and the yellow arrows the location for the lateral measurements. Figure 101 (e) shows the mounting of the TMD's to the base using a M16 bolt into the steel support structure that was also used to mount the main bearings and hopper.

8.1 Results

Results of the sound intensity scans over the decanter are shown in Figure 102. The scans were undertaken by the method described in Section 2.7. The two scans produced very consistent results, less than 0.6 dB variation, except for the two lowest and highest frequency bands and the 400 Hz one-third octave bands. Therefore the scans are deemed to be a fair representation of the sound that the decanter produces. The overall sound power of the decanter was 100.8 dB.

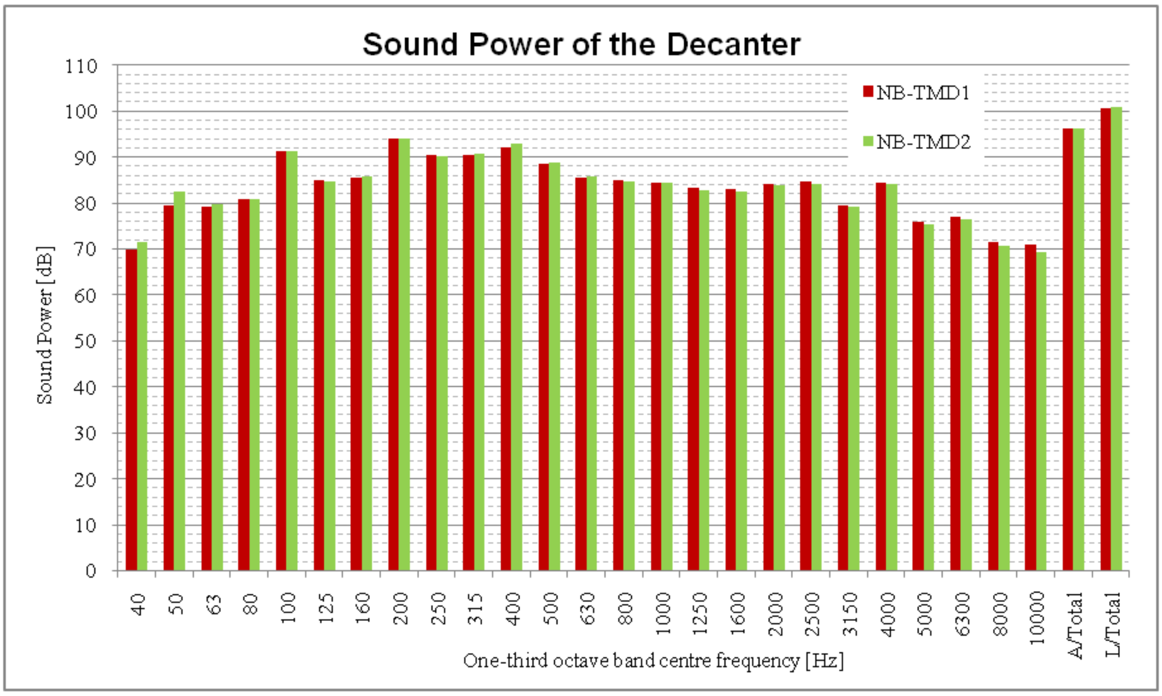


Figure 102 Sound power of the decanter

After the intensity scans were completed the natural frequencies of the TMD's were re-assessed and the results are shown in Figure 103. The first configurations, with the TMD's tuned to 54 Hz, are A – B and D – E. The natural frequency remained at 54 Hz and the second natural frequency in the lateral direction was still just above 108 Hz, at 114 Hz. The natural frequency of the second configuration is shown in C and F. The natural frequency of TMD C has decreased slightly to 98 Hz but has a second natural frequency at 108 Hz. The natural frequency of TMD F has remained at 108 Hz.

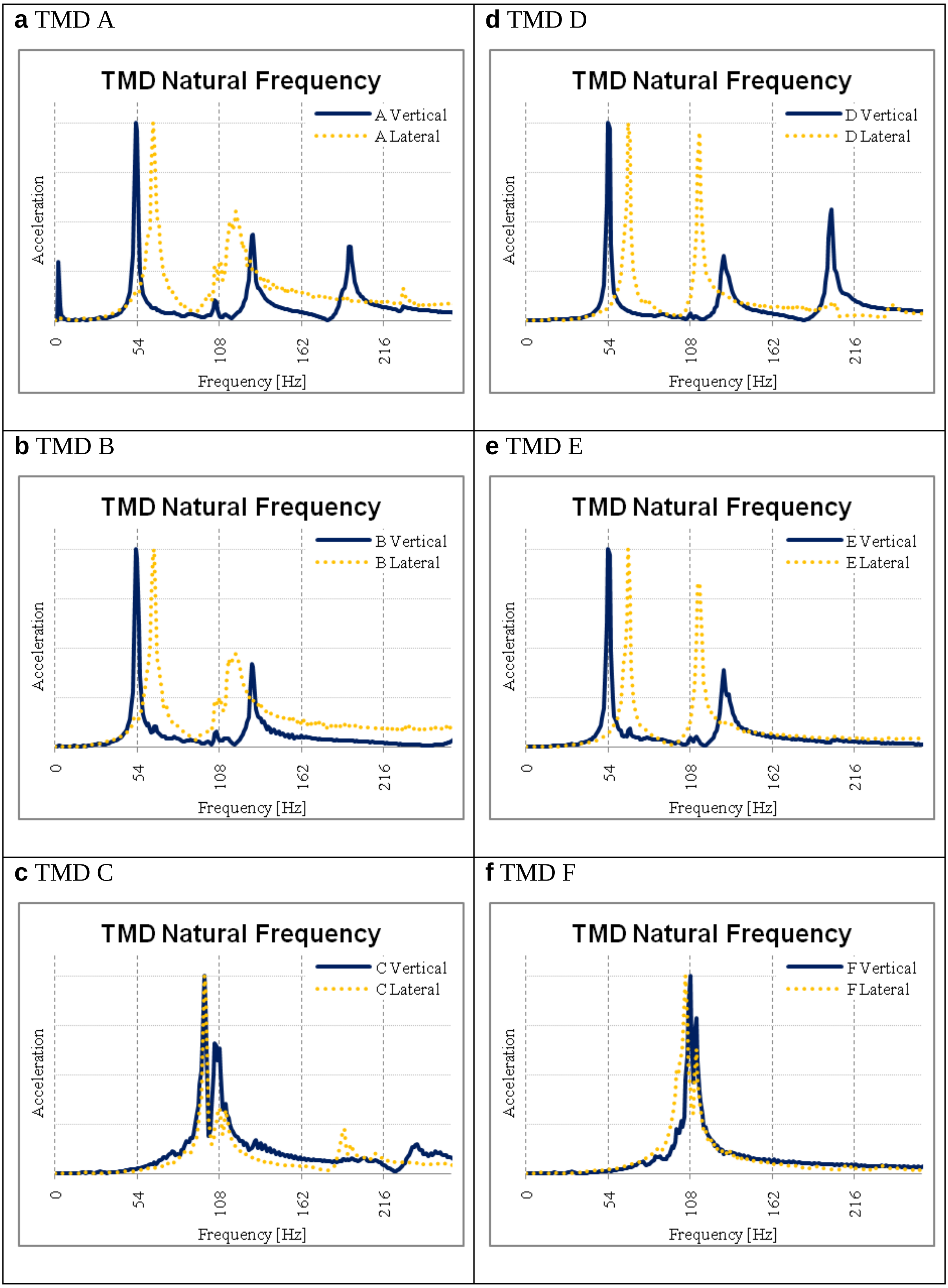


Figure 103 (a-f) Natural frequencies of the tuned mass dampers

Vibration acceleration measurements were taken after the re-evaluation of the natural frequencies of the TMD's was completed. The overall results for the acceleration measurements are shown in Table 21 and the spectrum in Figure 104. The previous configurations measurements are presented in Sections 3.1.3, 5 and 7. The measurements showed a small decrease on the base, a small increase slightly on the hopper, and an unchanged level on the gearbox guard. There was a significant decrease in the overall accelerations measured at 108 Hz. There was also a reduction at 216 Hz but most of the other lower harmonics showed a slight increase.

Table 21 Overall acceleration results

	RMS Average Accelerations [m/s^2]			
Configuration	Base	Hopper	GB Guard	Total
Original	0.15	0.46	1.26	0.70
New Base	0.11	0.25	0.68	0.37
NB - HI	0.14	0.15	0.08	0.13
NB -TMD	0.12	0.16	0.08	0.13

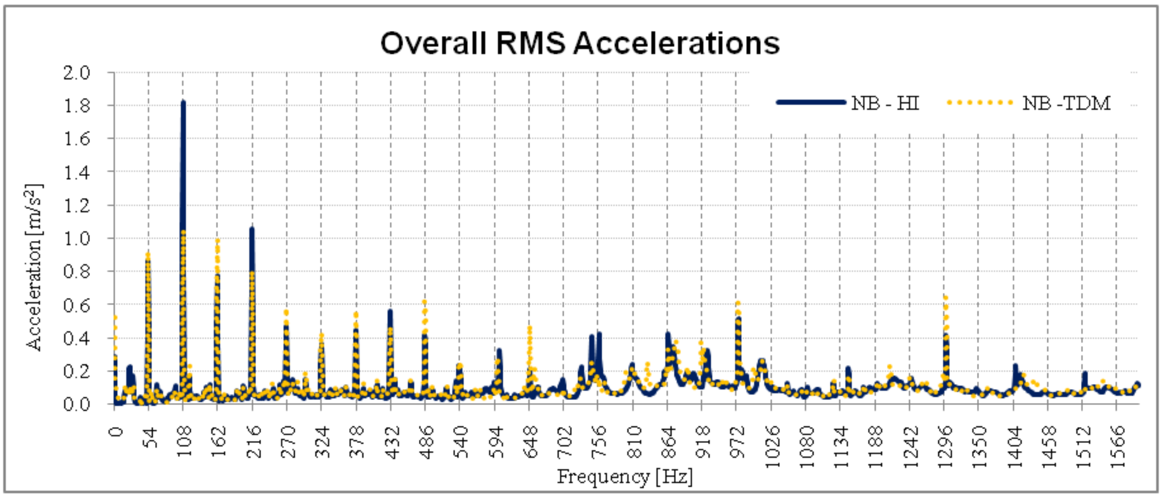


Figure 104 Total averaged RMS accelerations of driven harmonic frequencies

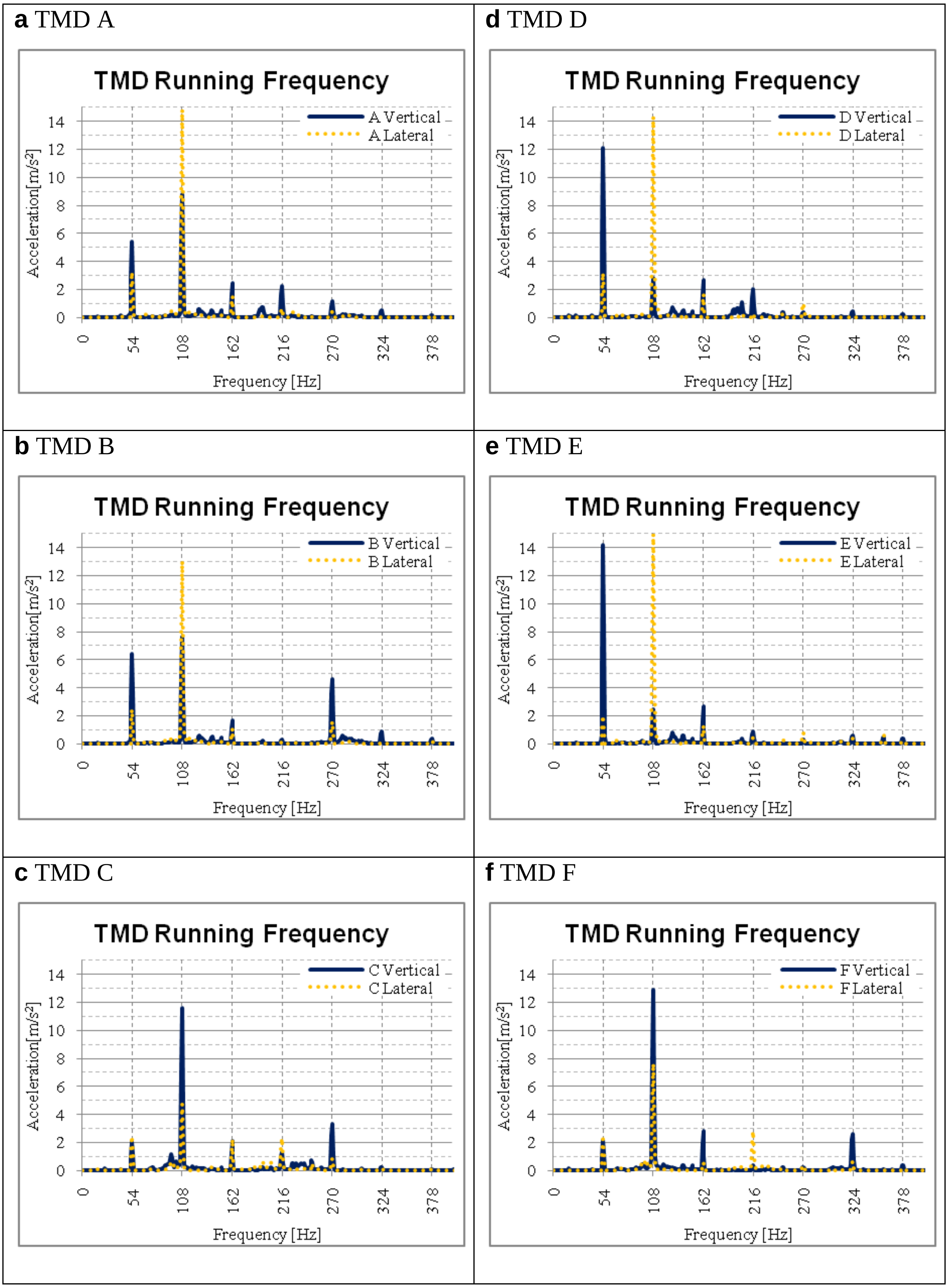


Figure 105 (a-f) Vibration measurements of the tuned mass dampers while decanter running

Once the standard acceleration measurement points were assessed while the decanter was running, the accelerations were measured on the TMD's. The results of the assessment are shown in Figure 105. The accelerations measured for 108 Hz ranged from 11.6 to 14.8 m/s2. TMD's D and E also had acceleration over 12 m/s2 for 54 Hz. The TMD's tuned primarily for 54 Hz had higher acceleration at 108 Hz than the two TMD's that were tuned for 108 Hz.

8.2 Comparison with Previous Configuration

8.2.1 Sound Intensity Measurements

The comparison of the sound intensity scans of the current decanter configuration, NB-TMD, and the previous configuration, NB-HI see Section 7, are shown in Figure 106 and Tables 22 to 24. The change in overall sound power due to the TMD's was less than 0.1 dB. The decanter produced the same overall sound power but in a slightly different manner.

Figure 106 and Table 22 show that the sound power of the decanter had no significant (more than 2 dB) changes in the one-third octave bands or regions. Reductions of over 1 dB occurred in the 50 Hz, 100 Hz and 125 Hz one-third octave bands with deductions of 1.6 dB, 1.7 dB and 1.6 dB respectively. Increases of over 1 dB occurred in the 315 Hz, 400 Hz, 500 Hz and 10,000 Hz one-third octave bands with increases of 1.2 dB, 1.4 dB, 1.3 dB and 1.8 dB respectively. The increase in the 10,000 Hz one-third octave band was not significant, due to it being at the extreme for measurements using the 12 mm spacer within the intensity probe, together with the low sound power level and will not be considered further.

There was a reduction in sound power in the 50 Hz and 100 Hz one-third octave bands (which also contain the two natural frequencies of the TMD's). Table 23 shows the sound intensity distribution in the 100 Hz one-third octave band. It shows that the reductions were from most regions of the decanter but the highest reductions were for the top surface.

The 400 Hz one-third octave band had the largest increase, 1.4 dB, and the sound intensity distribution is shown in Table 24. The table shows that the increases are largest through the front and back surfaces of the decanter but the increases cannot be attributed to a particular component.

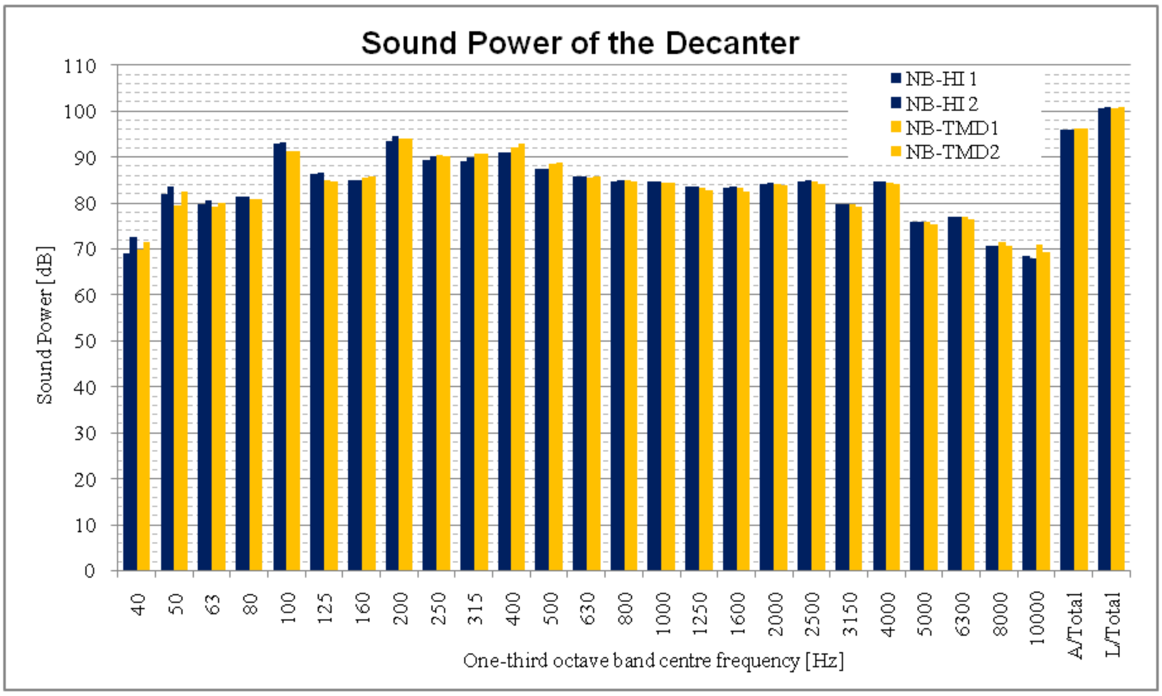


Figure 106 Sound power comparison

Table 22 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-HI1	84.0	86.7	87.1	80.1		90.0	92.8	92.0	85.4	89.3
NB-HI2	85.5	88.0	87.7	79.0		91.5	93.2	92.5	85.8	90.0
NB-TMD1	84.6	88.6	87.9	80.3		90.2	92.8	92.3	85.6	89.7
NB-TMD2	85.2	88.9	88.6	80.8		89.9	92.4	92.0	84.7	89.5
Ave Diff.	⇨ 0.2	⇧ 1.4	⇨ 0.8	⇨ 1.0		⇨ -0.7	⇨ -0.4	⇨ -0.1	⇨ -0.4	⇨ 0.0
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-HI1	82.9	87.8	87.6	85.3		84.2	92.1	91.7	87.0	88.9
NB-HI2	83.1	88.3	87.9	85.6		84.5	92.6	91.7	87.2	89.2
NB-TMD1	82.7	88.6	88.1	85.3		84.3	92.4	91.4	87.9	89.1
NB-TMD2	81.9	88.4	88.6	85.3		85.5	92.7	92.0	88.4	89.5
Ave Diff.	⇨ -0.7	⇨ 0.5	⇨ 0.6	⇨ -0.1		⇨ 0.6	⇨ 0.2	⇨ 0.0	⇧ 1.1	⇨ 0.2
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-HI1	86.3	88.1	87.4	89.4	88.6			83.2	86.5	85.3
NB-HI2	86.2	88.3	88.4	90.4	89.4			82.7	86.8	85.4
NB-TMD1	86.1	87.4	87.2	89.2	88.3			83.5	86.2	85.2
NB-TMD2	85.5	87.4	87.6	89.1	88.3			84.1	86.4	85.5
Ave Diff.	⇨ -0.5	⇨ -0.8	⇨ -0.5	⇨ -0.8	⇨ -0.7			⇨ 0.8	⇨ -0.3	⇨ 0.0
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-HI1	85.5	86.9	86.8	85.7	86.4					88.4
NB-HI2	86.5	87.3	87.0	85.6	86.7					88.9
NB-TMD1	85.3	87.2	86.8	86.3	86.6					88.6
NB-TMD2	85.4	87.7	86.8	86.4	86.8					88.7
Ave Diff.	⇨ -0.7	⇨ 0.4	⇨ -0.1	⇨ 0.7	⇨ 0.2					⇨ 0.0

Table 23 Comparison of sound intensity measurements for the 100 Hz one-third octave band [dB]

One-third octave band centre frequency:				100	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-HI1	71.4	78.0	75.0	62.8		80.5	86.7	84.8	78.2	81.9
NB-HI2	75.0	78.3	76.0	0.0		83.7	87.2	85.4	77.8	82.6
NB-TMD1	73.8	79.6	77.2	65.9		81.7	85.6	83.5	76.2	81.3
NB-TMD2	70.3	75.2	74.7	64.3		77.3	81.5	80.3	73.4	77.5
Ave Diff.	-1.2	-0.8	0.5	2.3		-2.6	-3.4	-3.2	-3.2	-2.9
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-HI1	70.7	75.3	74.8	74.3		79.4	85.4	85.4	79.6	81.5
NB-HI2	71.6	76.7	76.2	74.4		79.1	85.3	85.2	78.6	81.5
NB-TMD1	68.7	75.9	76.1	72.1		76.6	82.5	83.3	78.5	79.4
NB-TMD2	67.3	76.7	76.9	73.3		79.9	85.4	85.4	79.8	81.7
Ave Diff.	-3.2	0.3	1.0	-1.7		-1.0	-1.4	-1.0	0.1	-0.9
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-HI1	80.0	82.8	81.2	81.6	81.7			68.8	79.1	76.7
NB-HI2	80.2	83.4	81.2	81.7	81.9			61.3	78.4	75.8
NB-TMD1	79.4	82.1	79.9	79.9	80.4			67.5	74.9	72.8
NB-TMD2	79.7	82.4	81.1	81.3	81.4			70.5	74.3	72.9
Ave Diff.	-0.5	-0.8	-0.7	-1.1	-0.9			4.0	-4.2	-3.4
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-HI1	73.6	77.2	75.4	58.7	74.7					80.6
NB-HI2	73.4	77.0	75.4	0.0	74.4					80.9
NB-TMD1	69.0	71.5	68.3	0.0	68.6					79.1
NB-TMD2	67.4	72.8	69.8	0.0	69.5					79.0
Ave Diff.	-5.3	-4.9	-6.4	-	-5.5					-1.7

Table 24 Comparison of sound intensity measurements for the 400 Hz one-third octave band [dB]

One-third octave band centre frequency:				400	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-HI1	72.4	78.0	79.5	74.7		76.8	81.2	82.1	75.8	79.1
NB-HI2	71.0	78.5	79.7	73.0		77.6	81.7	82.2	75.8	79.3
NB-TMD1	72.8	81.2	81.1	71.9		74.5	82.6	82.7	77.2	80.3
NB-TMD2	75.2	82.7	83.3	74.7		75.4	83.9	83.3	76.6	81.6
Ave Diff.	2.3	3.7	2.6	-0.5		-2.2	1.8	0.8	1.1	1.7
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-HI1	74.8	80.1	79.4	75.9		77.0	82.0	82.5	77.0	79.9
NB-HI2	73.5	79.4	78.6	76.2		76.7	81.8	82.1	77.1	79.5
NB-TMD1	73.0	79.7	80.9	74.6		76.7	83.7	83.8	77.8	80.8
NB-TMD2	70.5	79.8	82.0	72.9		77.7	84.6	84.4	78.8	81.5
Ave Diff.	-2.4	0.0	2.5	-2.3		0.4	2.3	1.8	1.3	1.5
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-HI1	77.2	79.4	75.7	77.5	77.8			74.0	76.1	75.3
NB-HI2	76.3	79.2	74.6	77.6	77.6			74.7	76.1	75.5
NB-TMD1	74.5	79.0	75.7	78.6	78.0			74.9	75.4	75.2
NB-TMD2	75.0	78.9	76.4	79.2	78.5			75.6	76.3	76.0
Ave Diff.	-2.0	-0.4	0.9	1.4	0.6			0.9	-0.3	0.2
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-HI1	74.4	79.5	79.7	80.1	79.2					79.0
NB-HI2	74.1	79.6	79.4	79.5	79.0					78.9
NB-TMD1	75.2	81.9	79.2	79.2	79.9					79.9
NB-TMD2	75.9	83.0	79.5	79.9	80.7					80.8
Ave Diff.	1.3	2.9	-0.2	-0.2	1.1					1.4

8.2.2 Acceleration Measurements

Figure 107 shows some individual vibration acceleration measurements of the base. The points were selected at each end of the base and in the middle. Both vertical and lateral measurement points were selected.

The high vertical acceleration measurements at the gearbox end, compared to those at the main drive end, are likely due to the cantilevered gearbox inducing higher accelerations and/or the mesh flooring of the test room allowing the gearbox end of the decanter to vibrate more freely. The changes in measured accelerations at the middle and gearbox end are consistent with the overall results for the base. This was due to the way that the averaging was done, with emphasis for higher values carrying into the averaged result. There was a measured reduction for the vertical and lateral measurement points at the main drive end, 46 % and 41 % respectively, which can be attributed to the high acceleration measurements for TMD C.

Figures 108 to 116 show the vibration acceleration measurements of the three main components under investigation on the decanter. The acceleration measurements have been combined by taking the ‘root of the mean of the squares’ (RMS) of the various individual measurements. The spacing of the frequency labels were chosen to match the harmonic frequencies of the rotating bowl.

The accelerations measured on the base, for 108 Hz and 216 Hz, showed a decrease of 40 % and 33 % respectively. This indicates that the high acceleration measurements at 108 Hz on the TMD’s correlate to a reduction in accelerations in the base for the same frequency. The gearbox guard also had a slight decrease at 108 Hz but there was no change in the acceleration measured for 108 Hz on the hopper. The base had minor increases in acceleration for most of the other harmonic frequencies. The gearbox guard and hopper had minor changes in vibration levels for various frequencies but the overall levels stayed the same. There was a 40 % increase in the level for hopper at 216 Hz.

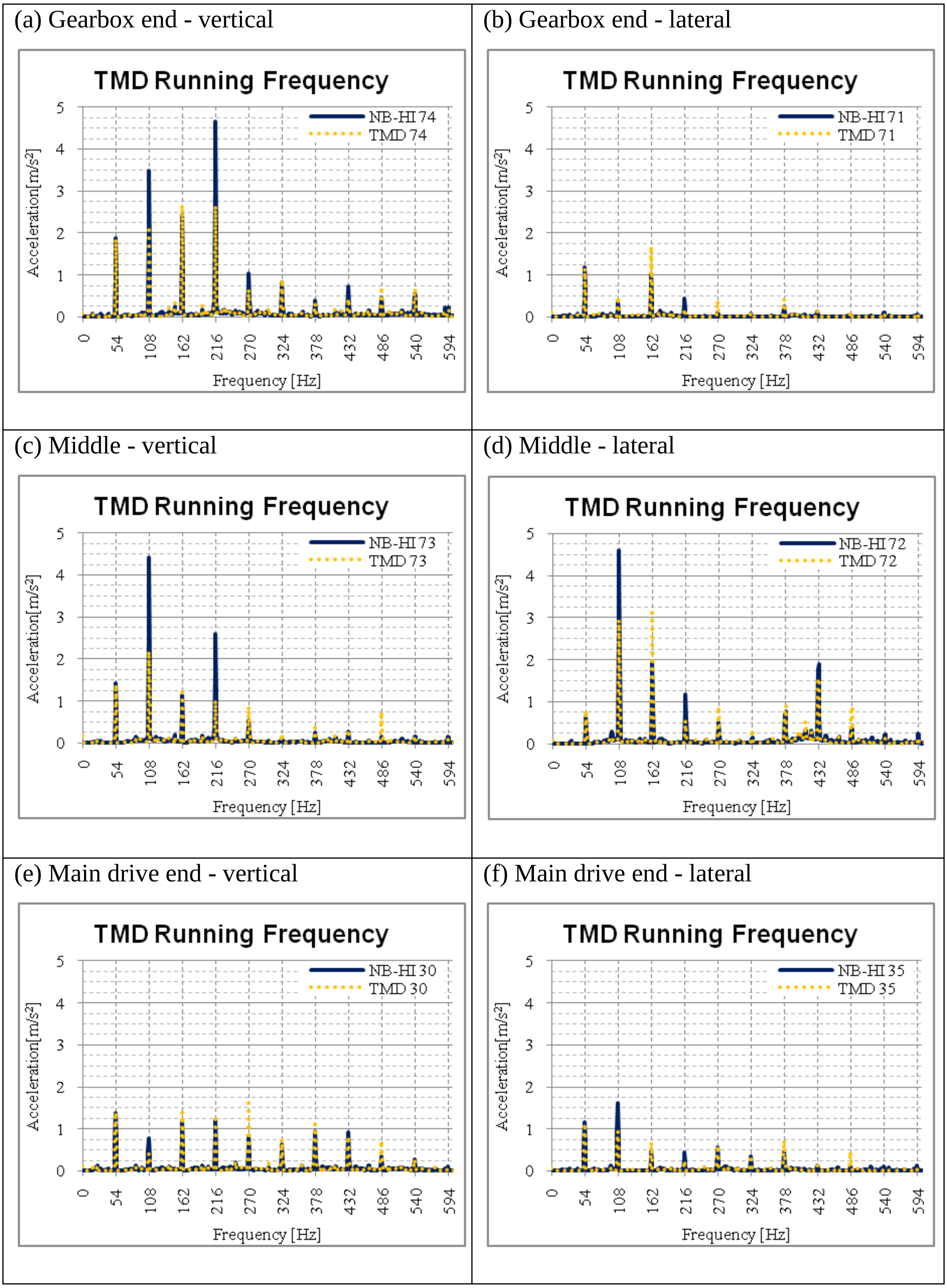


Figure 107 (a-f) Base accelerations at individual locations

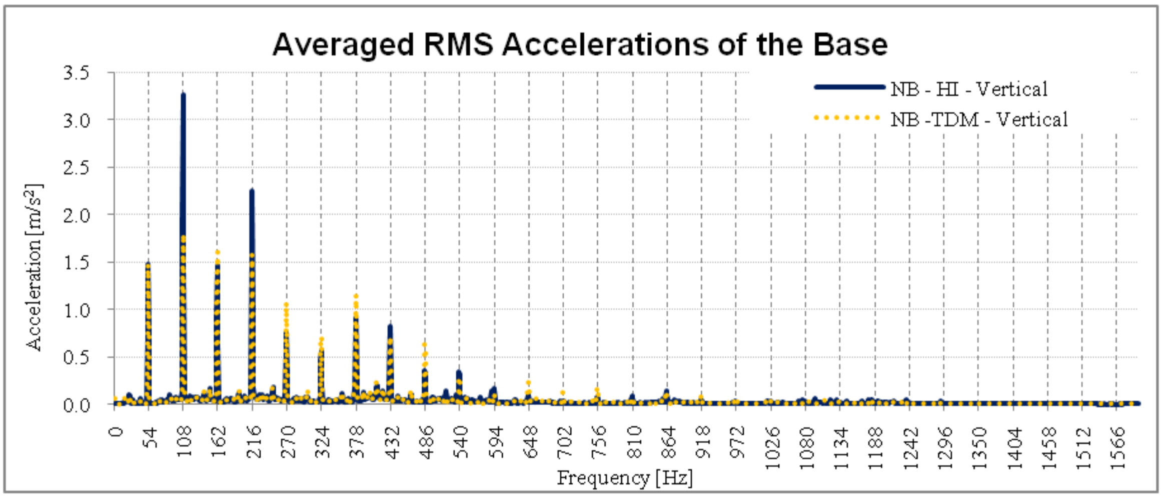


Figure 108 Accelerations of the base – vertical

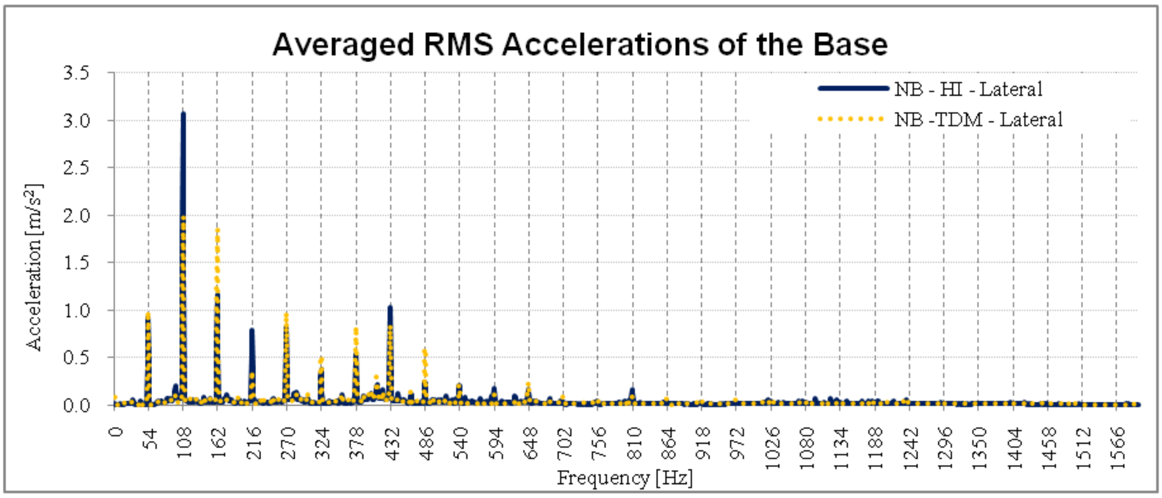


Figure 109 Accelerations of the base – lateral

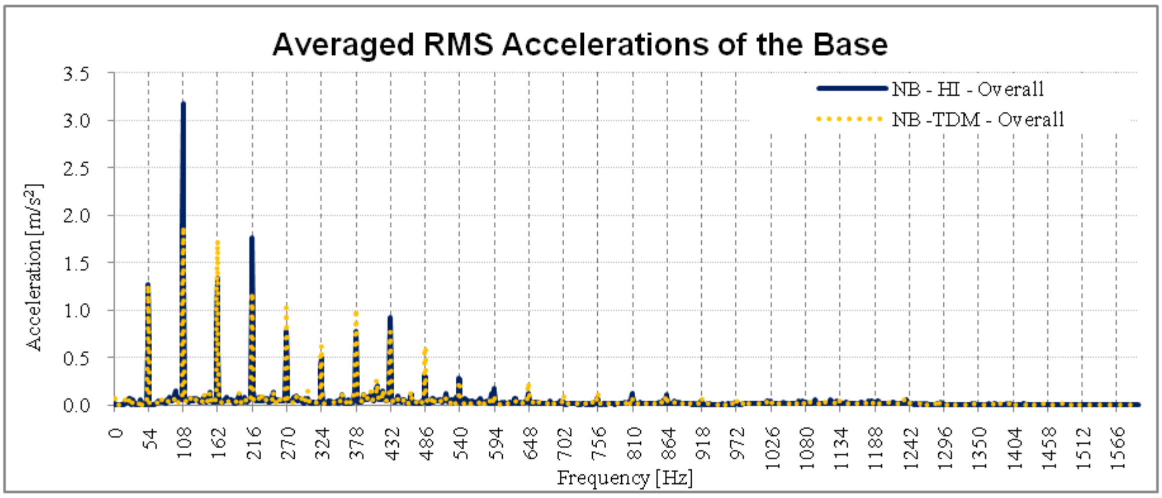


Figure 110 Accelerations of the base – overall

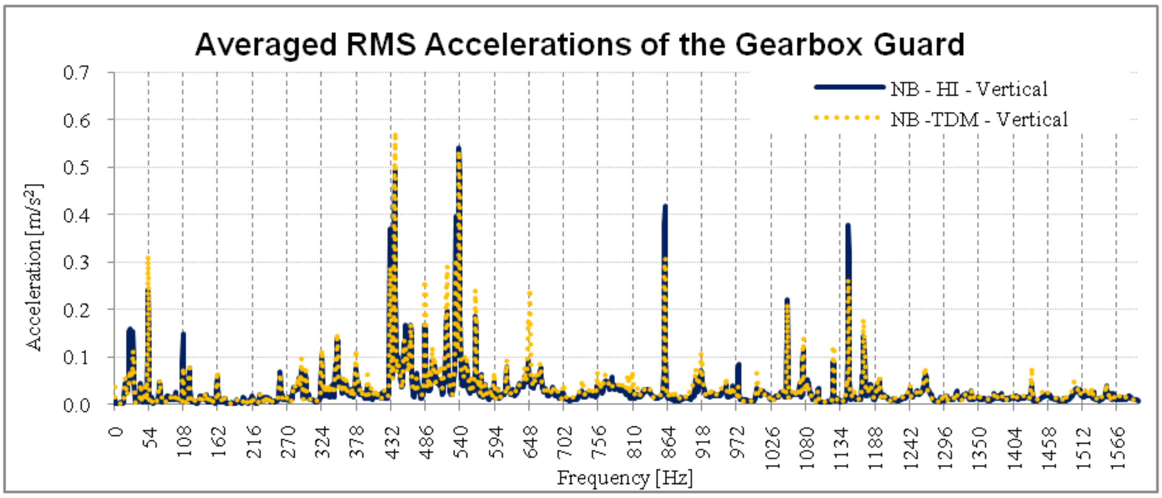


Figure 111 Accelerations of the gearbox guard – vertical

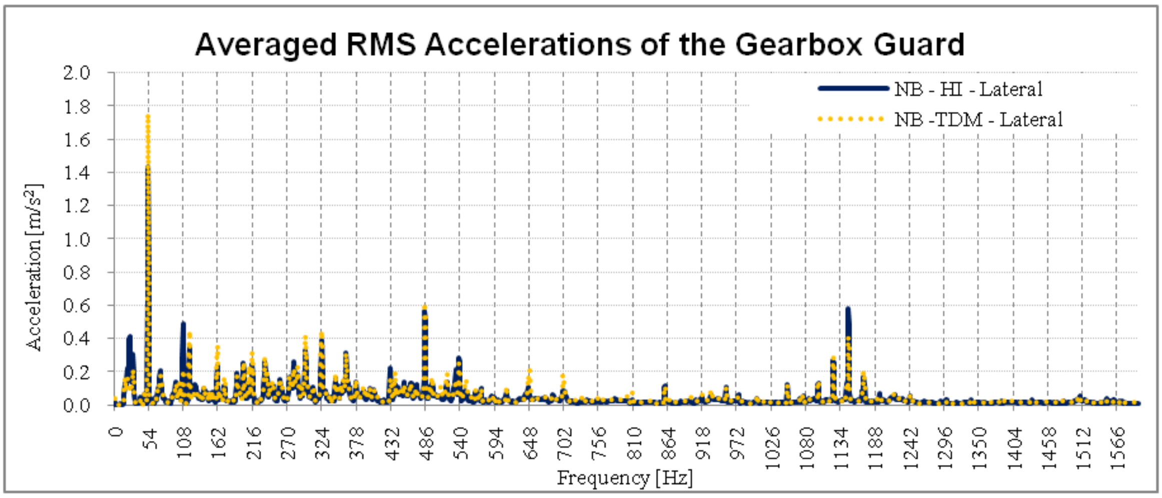


Figure 112 Accelerations of the gearbox guard – lateral

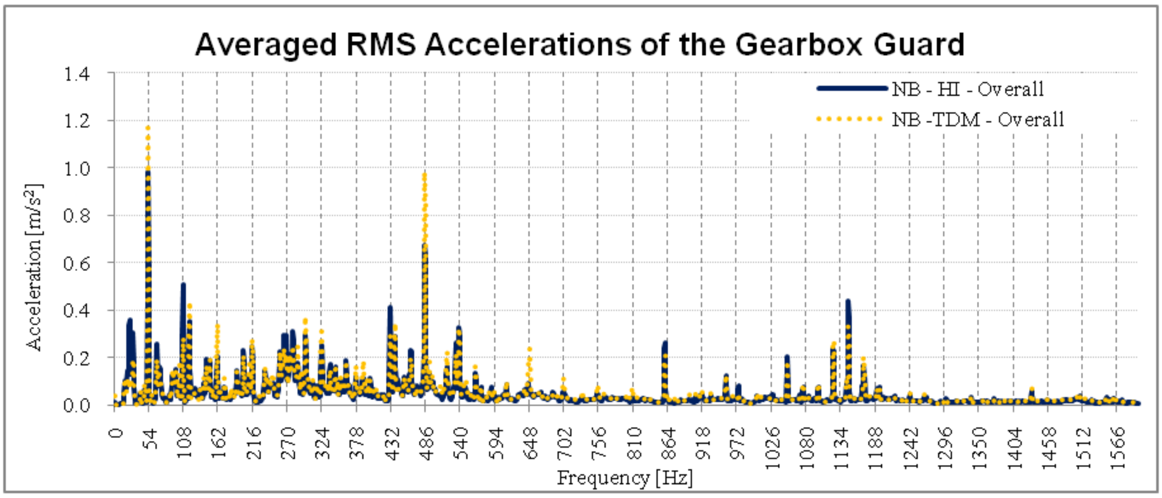


Figure 113 Accelerations of the gearbox guard – overall

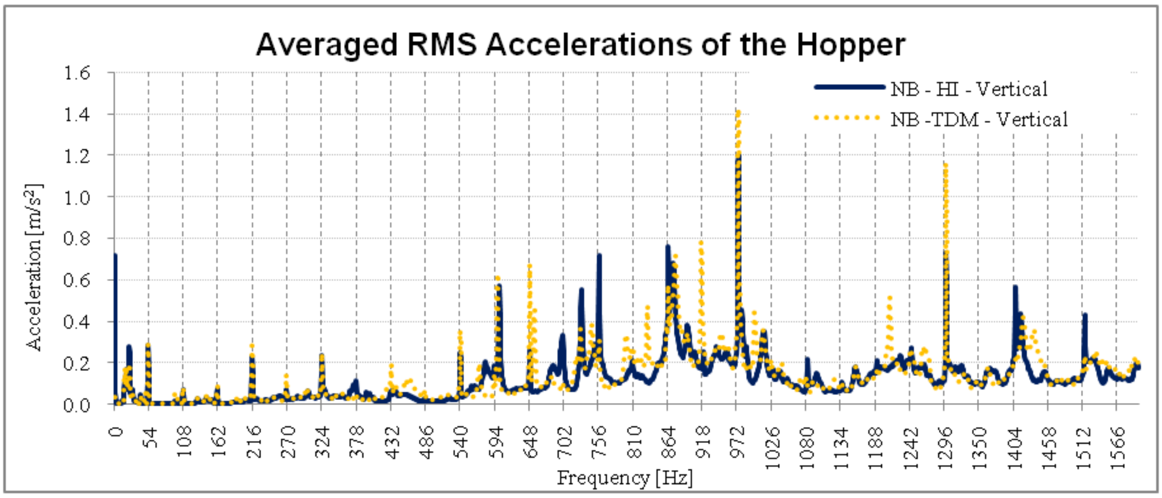


Figure 114 Accelerations of the hopper – vertical

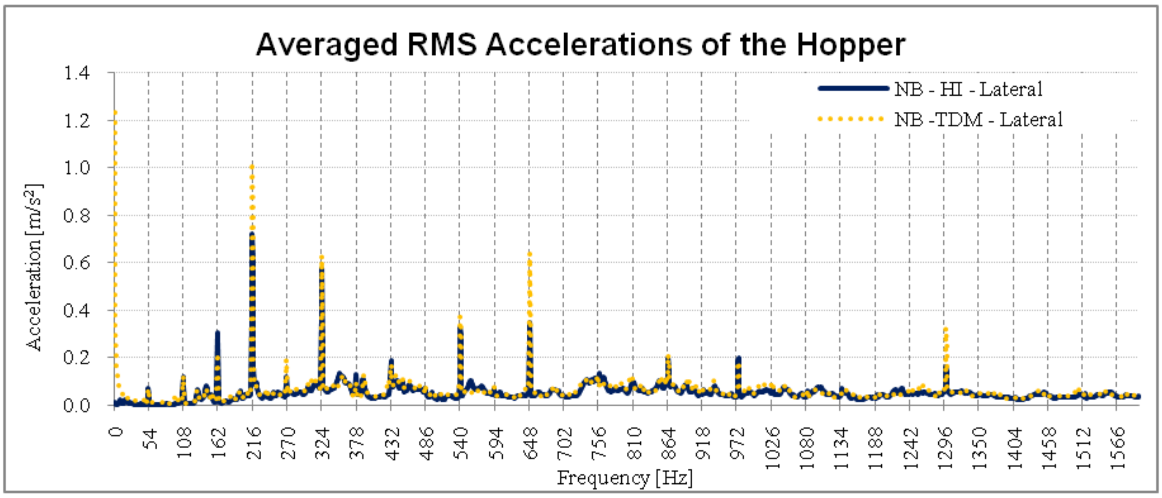


Figure 115 Accelerations of the hopper – lateral

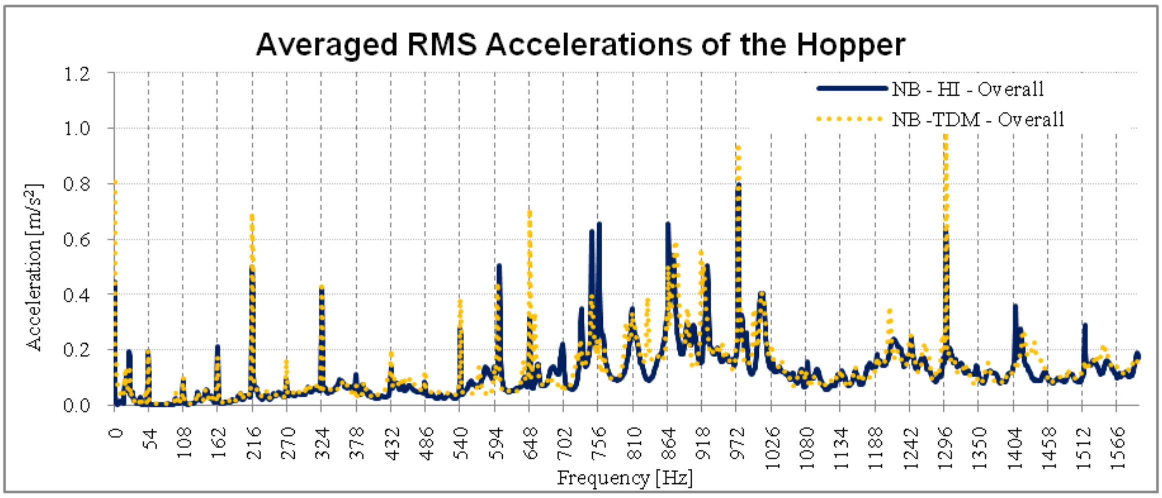


Figure 116 Accelerations of the hopper – overall

8.3 Comparison with original configuration

Comparisons of the current configuration and the original configuration (OC), see Section 3.1.3 are shown in Figures 117 to 120 and Tables 21, 25 and 26. The changes to the decanter resulted in a 3.3 dB reduction in sound power. There were significant reductions in sound power in the one-third octave bands of 50 Hz to 160 Hz (excluding 80 Hz) and 5,000 Hz and above. There was a 1.8 dB increase in sound power in the 200 Hz one-third octave band and a 2.5 dB decrease in the 250 Hz one-third octave band. The remaining higher one-third octave bands had less than a 2 dB change in sound power level.

Table 25 shows that the decrease came from all regions of the decanter except for the lower regions of the front and back surfaces. All the regions directly associated with the base, gearbox guard and hopper showed significant reductions, with reductions of 2.8 dB to 10 dB.

The one-third octave band with the highest sound power level was 200 Hz. Here there was a 1.8 dB increase from the original configuration. Table 26 shows the distribution of the 200 Hz one-third octave band – the regions with significant increases, with sound intensity levels over 84 dB, are from the lower regions of the front, back and left surfaces. These surfaces are not directly associated with base, gearbox guard or hopper.

The comparison in acceleration measurements are shown in Table 21, and the spectrums are shown in Figures 118 to 120. The acceleration measurements for the base increased at 216 Hz and 486 Hz and did not change at 108 Hz. The acceleration measurements at the remaining frequencies decreased as did the overall measurements. The acceleration measurements for the gearbox guard and hopper decreased significantly for all frequencies, except at 1296 Hz on the hopper.

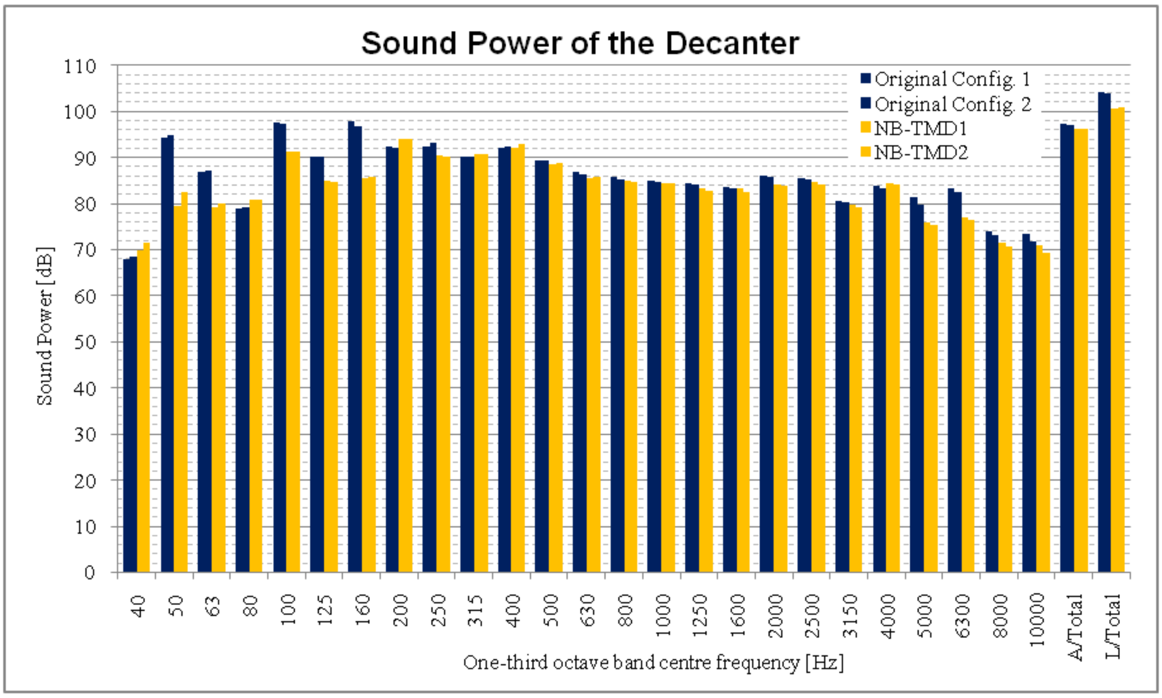


Figure 117 Sound power comparison

Table 25 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	92.7	91.4	91.9	87.9		89.4	91.5	93.0	87.1	91.1
OC2	93.1	91.6	93.0	88.2		89.4	91.7	93.1	87.4	91.5
NB-TMD1	84.6	88.6	87.9	80.3		90.2	92.8	92.3	85.6	89.7
NB-TMD2	85.2	88.9	88.6	80.8		89.9	92.4	92.0	84.7	89.5
Ave Diff.	⇩ -8.0	⇩ -2.8	⇩ -4.2	⇩ -7.5		⇨ 0.7	⇨ 1.0	⇨ -0.9	⇩ -2.1	⇩ -1.7
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	86.7	92.5	93.0	96.0		75.9	91.0	91.2	91.0	91.4
OC2	87.4	92.5	92.9	95.1		80.7	91.7	91.4	90.3	91.4
NB-TMD1	82.7	88.6	88.1	85.3		84.3	92.4	91.4	87.9	89.1
NB-TMD2	81.9	88.4	88.6	85.3		85.5	92.7	92.0	88.4	89.5
Ave Diff.	⇩ -4.8	⇩ -4.0	⇩ -4.6	⇩ -10.3		⇧ 6.6	⇧ 1.2	⇨ 0.4	⇩ -2.5	⇩ -2.1
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	93.4	97.8	96.2	91.9	94.6			89.9	90.5	90.2
OC2	93.8	97.0	95.5	92.0	94.2			88.9	89.6	89.3
NB-TMD1	86.1	87.4	87.2	89.2	88.3			83.5	86.2	85.2
NB-TMD2	85.5	87.4	87.6	89.1	88.3			84.1	86.4	85.5
Ave Diff.	⇩ -7.8	⇩ -10.0	⇩ -8.4	⇩ -2.8	⇩ -6.1			⇩ -5.6	⇩ -3.7	⇩ -4.4
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	94.5	91.3	93.9	90.2	92.6					91.9
OC2	93.7	92.8	92.1	88.7	92.0					91.8
NB-TMD1	85.3	87.2	86.8	86.3	86.6					88.6
NB-TMD2	85.4	87.7	86.8	86.4	86.8					88.7
Ave Diff.	⇩ -8.8	⇩ -4.6	⇩ -6.2	⇩ -3.1	⇩ -5.6					⇩ -3.2

Table 26 Comparison of sound intensity measurements for the 200 Hz one-third octave band [dB]

One-third octave band centre frequency:				200	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
OC1	82.0	80.5	79.8	76.2		81.5	79.8	79.9	75.9	79.7
OC2	82.5	81.1	78.7	75.4		82.9	81.6	81.6	76.4	80.5
NB-TMD1	77.0	80.0	77.7	0.0		86.7	86.6	85.4	80.1	83.1
NB-TMD2	80.0	81.8	77.9	0.0		86.8	86.7	85.0	79.1	83.2
Ave Diff.	-3.8	0.1	-1.4	-		4.6	6.0	4.5	3.4	3.1
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
OC1	73.0	78.2	79.5	82.3		0.0	82.9	82.1	83.8	80.2
OC2	71.1	78.9	77.9	81.2		0.0	82.3	81.3	80.7	79.3
NB-TMD1	0.0	81.8	80.5	79.4		0.0	86.9	83.6	82.5	82.3
NB-TMD2	0.0	80.9	81.9	80.4		0.0	86.3	84.4	82.4	82.3
Ave Diff.	-	2.8	2.5	-1.8		-	4.0	2.3	0.2	2.6
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
OC1	79.1	82.0	81.3	80.5	80.8			67.8	80.7	78.2
OC2	79.5	81.9	81.3	79.5	80.4			66.5	79.3	76.8
NB-TMD1	77.1	76.3	82.5	85.1	83.3			73.4	78.7	77.0
NB-TMD2	72.5	74.7	82.0	84.5	82.6			76.4	80.1	78.8
Ave Diff.	-4.5	-6.5	1.0	4.8	2.3			7.8	-0.6	0.4
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
OC1	82.9	78.9	82.2	78.4	80.8					80.1
OC2	81.6	82.8	78.4	78.0	80.6					79.9
NB-TMD1	79.8	76.3	78.0	79.5	78.4					81.9
NB-TMD2	79.6	76.9	77.8	79.5	78.4					81.9
Ave Diff.	-2.6	-4.3	-2.4	1.3	-2.3					1.9

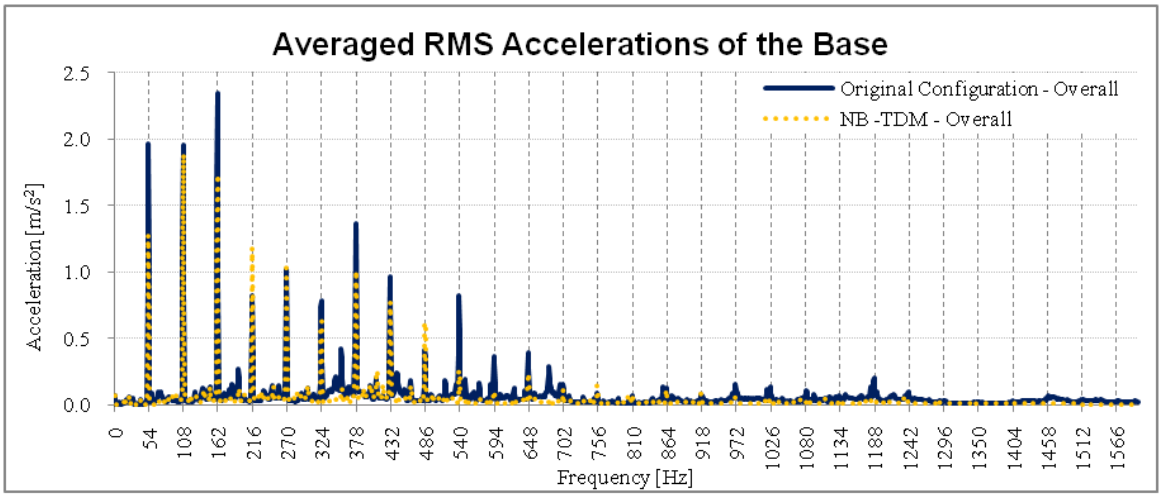


Figure 118 Accelerations of the base – overall

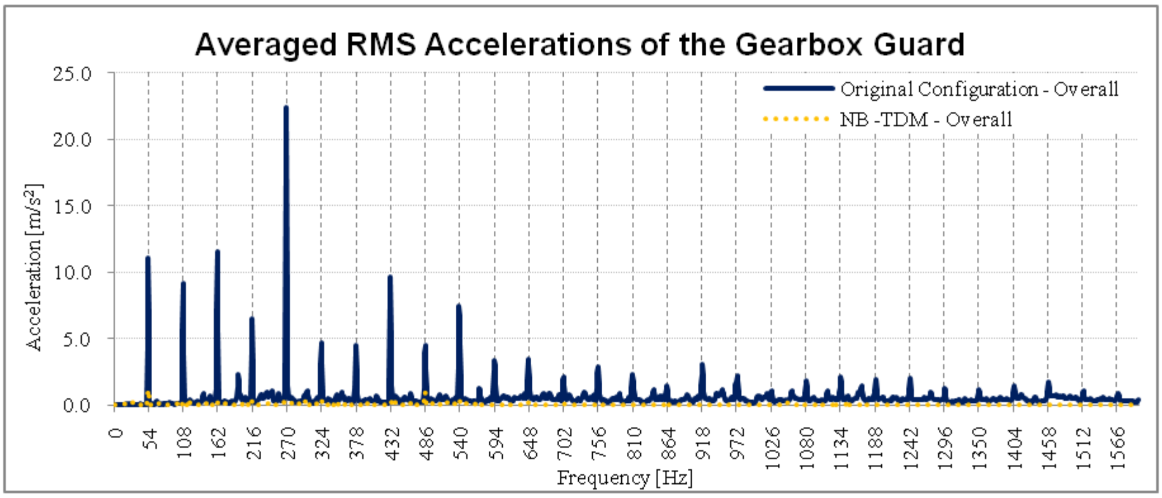


Figure 119 Accelerations of the gearbox guard – overall

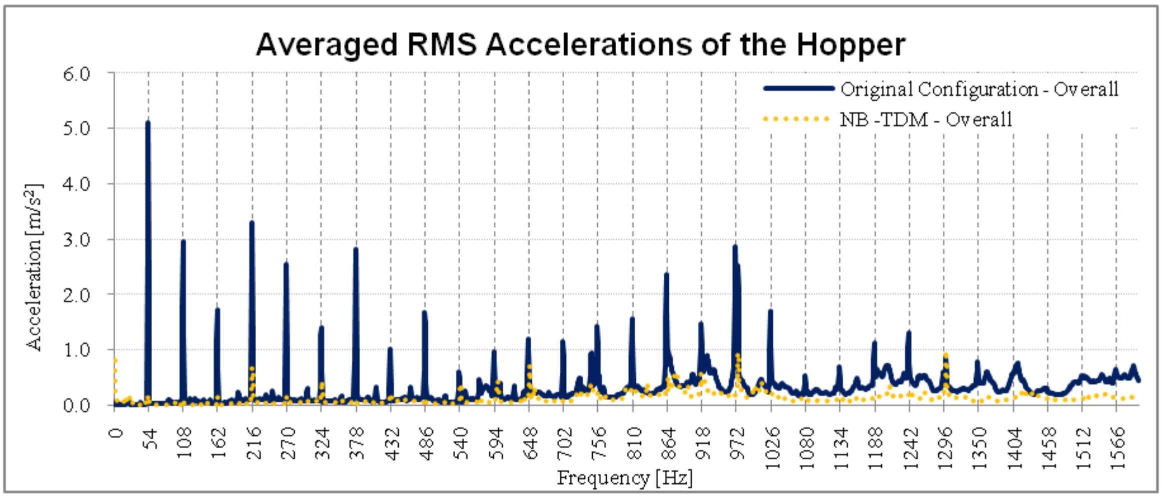


Figure 120 Accelerations of the hopper – overall

8.4 Discussion

8.4.1 The Effects of the Tuned Mass Dampers

The purpose of adding the TMD's was to reduce vibrations within the base and hence the decanter. The TMD's were to be tuned for 108 Hz and 216 Hz, the two one-third octave bands with the highest sound power in for the NB-HI configuration. Due to the supports for the TMD's not being rigid enough the TMD's were subsequently tuned to 54 Hz and 108 Hz. The addition of the TMD's resulted in the maximum recorded acceleration reducing from 5.0 m/s^2 to 3.2 m/s^2.

The tuned mass dampers mainly vibrated at 108 Hz and secondly at 54 Hz. There was no reduction in the vibrations measured on the base at 54 Hz. There was a minor reduction in sound power for the 50 Hz one-third octave band but the reduction was within the expected variation in measurement for this one-third octave band.

There was a 40 % reduction in measured acceleration for the base at 108 Hz (there was also a slight decrease in the gearbox guard for this frequency). The reduction in vibrations was also mirrored with a 1.7 dB reduction in sound power for the 100 Hz one-third octave band. This indicates that the addition of the TMD's resulted in a 1.7 dB reduction in the 100 Hz one-third octave band.

Though the TMD's were not producing any significant vibration levels at 216 Hz, the fourth harmonic of the rotating bowl, there was a 33 % reduction in vibrations measured at this frequency for the base. There was a 40 % increase in vibration at 216 Hz on the hopper. There was no change in sound power for the 200 Hz one-third octave band. This indicates the sound power reductions due to the base vibrating less at 216 Hz has been offset by increases in vibrations of the hopper.

The 1.7 dB reduction in sound power for the 100 Hz one-third octave band did not result in a reduction in the overall sound power of the decanter. This was because there was a 1.4 dB increase in the 400 Hz one-third octave band. The changes in sound power levels of the two bands has resulted in the 400 Hz one-third octave band containing the second highest sound power levels. The increase in sound power for the 400 Hz one-third octave band correlates to the top half of the decanter, towards the gearbox end, which encompasses all three components under investigation. There were some increases and decreases of the three components in the acceleration measurements for frequencies within the 400 Hz one-third octave band. As the vibration measurements do not indicate that a particular part of the decanter was responsible for the increase in sound power, it was likely that all three played a part in the increase.

8.4.2 The effects of reducing vibration levels

The focus of this and previous configurations has been on reducing the vibration levels within the decanter. Three components were identified as major contributors to the sound power of the decanter. They were the base, gearbox guard and hopper. The vibrations within the base have been reduced by changing to a polymer concrete base and adding TMD's. The vibrations in the gearbox guard and hopper were reduced by isolating these components from the base. The net effect of these modifications has been a moderate reduction of vibrations within the base and significant reductions for the gearbox guard and hopper.

The significant decreases in vibration levels have only resulted in a 3.3 dB reduction in sound power. All the significant reductions in sound power occurred in the 160 Hz one-third octave band and below and in the 5,000 Hz one-third octave band and above. The modifications significantly reduced the sound power in the three highest one-third octave bands, 50 Hz, 100 Hz and 160 Hz, for the original configuration. For the current configuration the 200 Hz and 400 Hz one-third octave bands have the highest sound power levels which have increased slightly from the original configuration.

There has been no significant change, 2 dB or over, in sound power level for the 315 Hz to 4,000 Hz one-third octave bands. This indicates that the sound power for these one-third octave bands was produced by a source other than the vibration of the base, gearbox guard and hopper.

The regions with the highest sound intensity, over 90 dB, are front and back, 6 and 7. The same regions also contain the highest sound intensity levels for the 200 Hz and 400 Hz one-third octave bands. The only components within these regions are part of the support frame and the back-drive motor. Neither of these components are likely sources for the majority of the sound. Therefore the sound was probably coming down from the lower surfaces of the decanter. A likely source for noise coming through the lower surface of the decanter was turbulence from the rotating bowl.

8.5 Conclusion

The TMD’s principally vibrated at 108 Hz. This resulted in a 40 % reduction in measured acceleration within the base and a 1.7 dB reduction in sound power for the 100 Hz one-third octave. The TMD’s were effective in reducing the sound power in the one-third octave band that correlate to the TMD’s principle vibrating frequency.

The TMD’s also had large accelerations at 54 Hz but these were not matched by decreases in measured accelerations at 54 Hz within the base. There was also no definitive sound power reduction for the 50 Hz one-third octave band.

The reductions in sound power in the 100 Hz one-third octave band were offset by increases in the 400 Hz one-third octave band. The addition of the TMD’s resulted in the maximum recorded acceleration reducing from 5.0 m/s^2 to 3.2 m/s^2. The net result of adding the TMD's was no change in the overall sound power of the decanter.

The reduction in vibration levels from the original decanter configuration to the current configuration has resulted in a 3.3 dB reduction in sound power. The reductions are primarily in the 160 Hz one-third octave band and below. There were only slight variations in sound power levels for the 315 Hz to 4,000 Hz one-third octave bands. Turbulence from the rotating bowl was a likely source of the sound power for these frequencies.

The following are possible options for the further reduction in the decanter sound power. It is recommended that:

- The air turbulence from the rotating bowl be investigated.
- The source of the sound power for the lower middle regions of the front and back be identified.

9 Modifications – Reducing Turbulence Noise

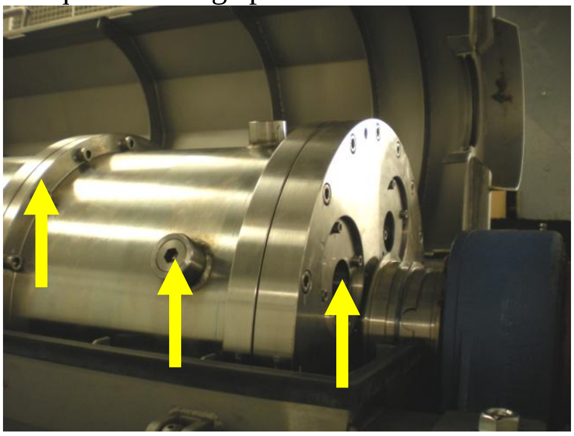
The decanter was modified in order to assess the turbulence noise component of the overall noise level. The rotating bowl assembly was considered the main source of turbulence noise. The decanter was modified in various ways to assess the contributions of difference parts of the decanter to the turbulence generated noise of the decanter. The three main areas of modification were:

- Modification A: Smoothing the outside of the bowl to reduce the turbulence due the various bowl protrusions.
- Modification B: Blocking the bowl exit ports to reduce the noise coming from the inside of the bowl and also the turbulence due to the bowl exit ports.
- Modification C: Blocking of the holes in the hopper that allows the noise from the bowl assembly direct access to the surrounding air.

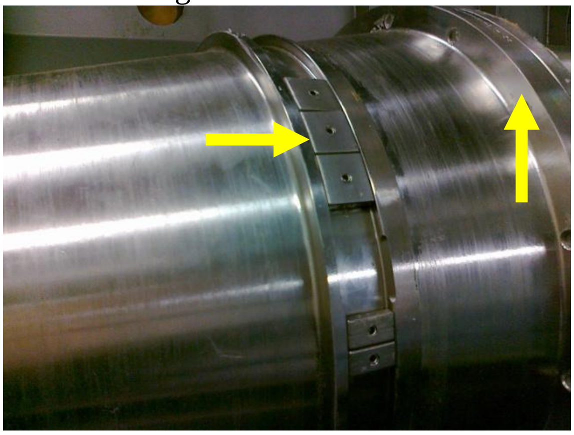
Figure 121 shows the decanter prior to modifications. Figure 121 (a) shows the liquid discharge ports of the bowl and the bolting of the bowl segments together. The discharge ports at this end of the bowl were not modified. The areas addressed in the modifications were turbulence due to the third phase liquid discharge ports on the side of the bowl and the bolt heads that hold the bowl segments together. Figure 121 (b) shows the balance ring and the bolt holes that were both addressed in the modifications. Figure 121 (c) shows the solids discharge ports in the bowl that were modified. Figure 121 (d) shows the hopper, which has eight holes and a discharge port, which were blocked off in the modifications. The discharge port is shown closed but was open in all prior testing of the decanter.

The modifications to smooth out the bowl's surface was achieved by using Selleys ® 'no more gaps' expanding foam and tape. The foam was used to fill in the gaps between the protrusions and the tape was used to support the foam under centrifugal loading. Various tapes were used such as duct, PVC, foil, and fibre reinforced. Due to the heat generated during the running of the decanter, the only tape that was effective was Scotch ® 893 fibre reinforced tape. Ados ® F2 adhesive was also used to hold down the ends of the tape.

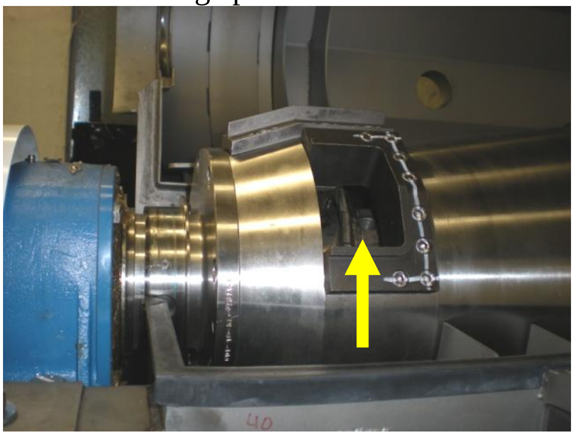
a Liquid discharge ports and bolt heads



b Balance ring and bolt holes



c Solid discharge port



d Holes within hopper

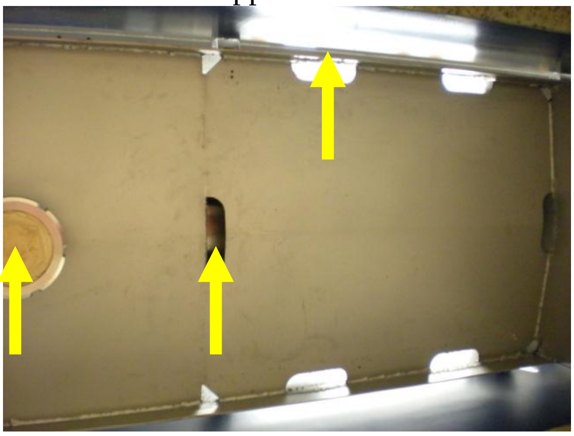


Figure 121 (a-d) Decanter prior to modifications

9.1 Decanter Modification One

9.1.1 The Modification – Smooth with Ports Covered

In the first modification, see Figure 122, the following changes were made:

- The gaps between the third phase liquid discharge ports were filled with foam and taped.
- The gaps between the bolts that hold the bowl segments together were filled with foam and taped.
- The bolt holes were filled with silicone gel.
- The balancing ring was taped over.
- The bowl at the solid discharge port was covered with foam and taped.
- The holes in the hopper were covered with 3 mm plate steel and sealed with silicone gel.
- The discharge port on the hopper was blocked with MDF.

- The third phase liquid and solid discharge ports were blocked.

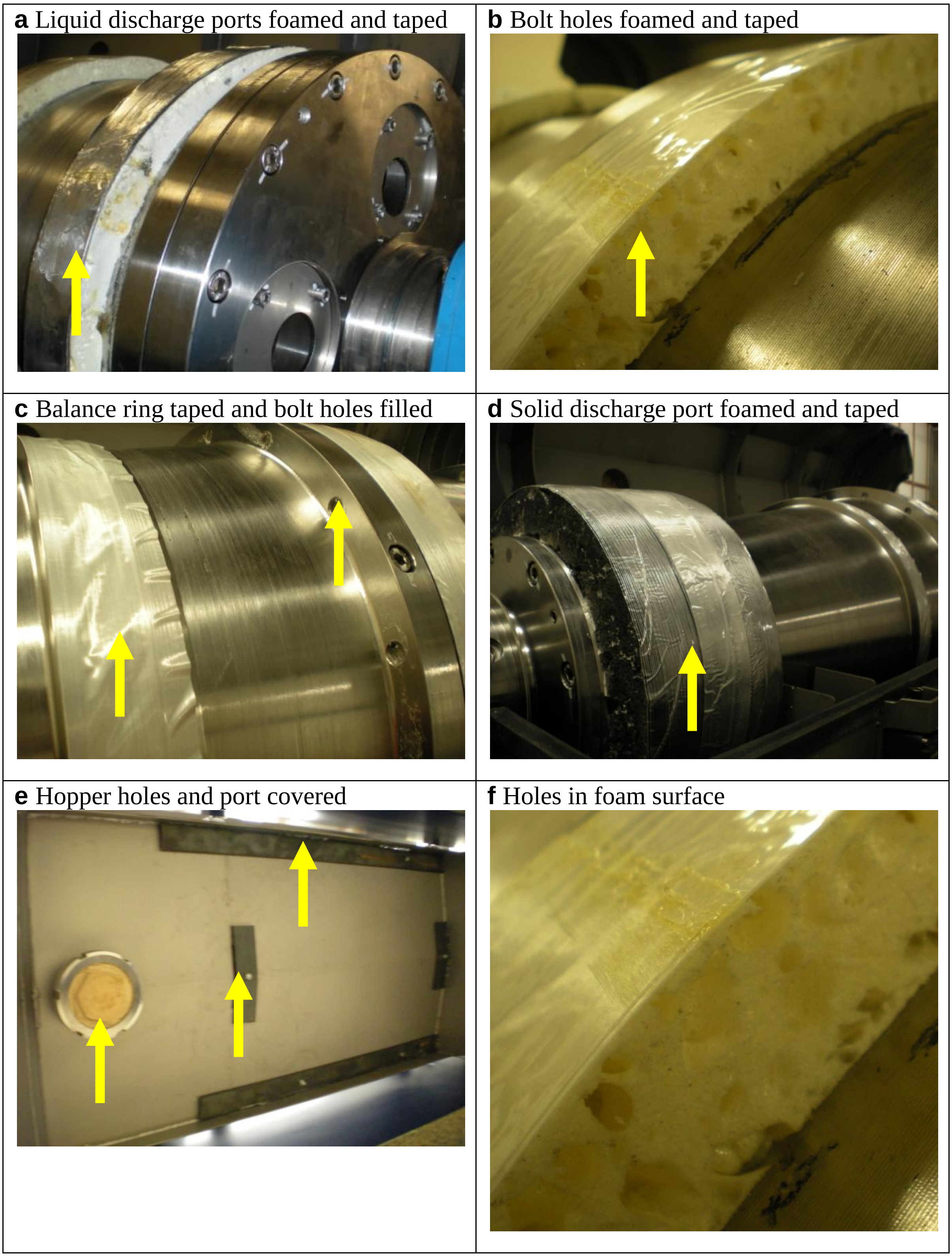


Figure 122 (a-e) Decanter modifications

9.1.2 Results of the Modification

The measured sound power of the decanter is shown in Figure 123. The scans were undertaken by the method described in Section 2.7. The results of scan 1 are not shown due to a failure of part of the modification that distorted the scan measurement. Except for the 40 Hz and 4,000 Hz one-third octave bands, the two scans produced consistent results. There was a requirement to modify the taping used in the modification between the two scans and this accounts for the minor variations in the levels recorded. The large variation at the 40 Hz one-third octave band does not impact the results and is only presented as an indicator of sound power levels below the 50 Hz one-third octave band. The large reduction at the 4,000 Hz one-third octave band appears to be an abnormal reading, and will be discussed later within this report. Therefore the scans were deemed to be a fair representation of the sound that the decanter produced. The overall sound power of the decanter was 95.1 dB.

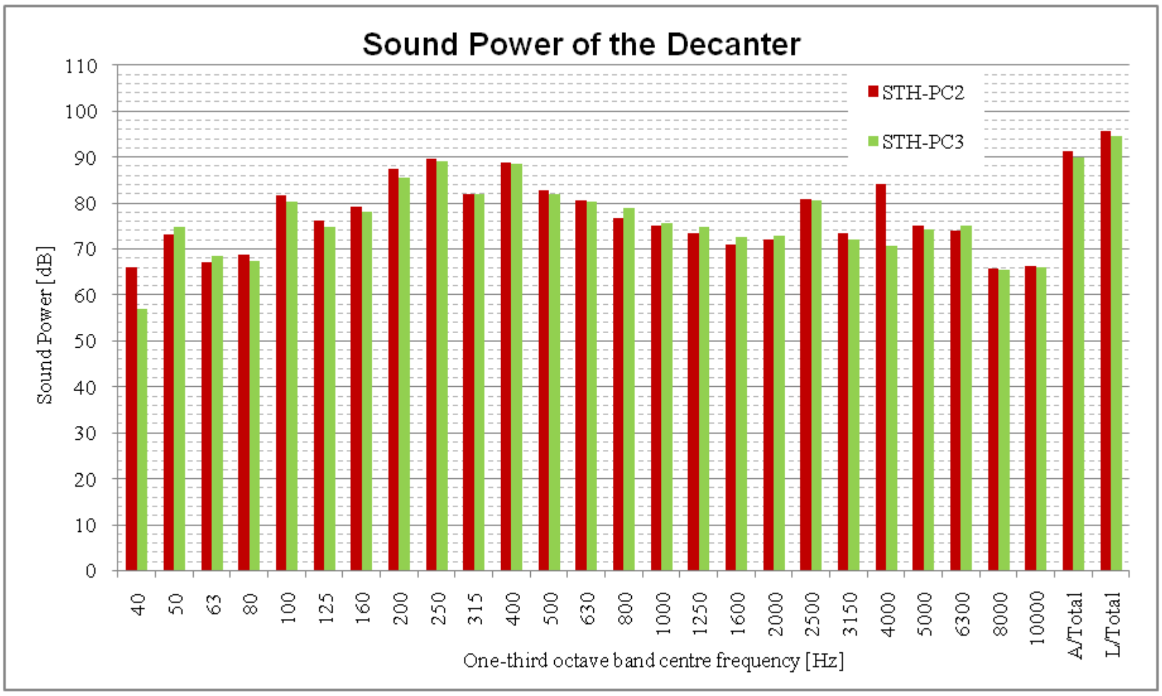


Figure 123 Sound power of the decanter

9.1.3 Comparison with NB-HI Configurations

The overall sound power of the current decanter configuration, STH-PC, and the NB-HI configuration, see Section 7.1, are shown in Figure 124 and Tables 27 to 30. The reduction in overall sound power due to smoothing the bowl and blocking the holes within the hopper was 5.6 dB. A 5.6 dB reduction represents a 70 % reduction in sound power.

Except at 250 Hz and 4,000Hz, Figure 124 shows that the sound power of the decanter reduced across all one-third octave bands. The reduction in sound power was from across all areas as shown in Table 28. The reduction in sound power for the various rotating bowl harmonics are shown in Table 27. The forth bowl harmonic relates to the solid discharge ports and the liquid discharge ports in the end of the bowl. The sixth bowl harmonic relates to the third phase liquid discharge ports on the side of the bowl. The sixteenth bowl harmonic relates to the bolt heads and bolt holes. The thirty-second harmonic relates to double the number of bolt heads and bolt holes. It was evident that the reduction in turbulence from the rotating bowl resulted in significant reductions in sound power, particularly for the one-third octave bands that contain bowl harmonic frequencies.

There was no reduction in sound power for the 250 Hz one-third octave band as a result of smoothing of the bowl. This indicates that turbulence noise was not the main noise source for this one-third octave band. Table 29 shows the sound intensity measurements for the 250 Hz one-third octave band. No region that had the highest sound intensity level. The main noise source for the 250 Hz one-third octave band cannot be determined from the current results but it was unlikely to be turbulence noise from the rotating bowl.

For the 4,000 Hz one-third octave band, see Table 30, the STH-PC3 scan produced sound intensity levels significantly lower than the other three scans. The eight subsequent scans to those used in Figure 123 and 11 previous scans gave a sound power level for the 4,000 Hz one-third octave band between 83.4 dB and 84.7 dB. These results indicate that the low level recorded for the 4,000 Hz one-third octave band in the STH-PC3 scan was abnormally low. The likely cause was another source external to the decanter producing noise within the 4,000 Hz one-third octave band. The highest level for the 4,000 Hz one-third octave band occurred in area Back 7. This area corresponds to the back-drive motor.

Table 27 Sound power reductions for the rotating bowl harmonic frequencies

Bowl Harmonic	Frequency [Hz]	One-Third Octave Band	Sound Power Level Reduction [dB]	Sound Power Reduction [%]
1	54	50	8.7	87
2	108	100	12.0	94
4	216	200	7.6	83
6	324	315	7.6	83
16	864	800	7.0	80
32	1728	1600	11.7	93

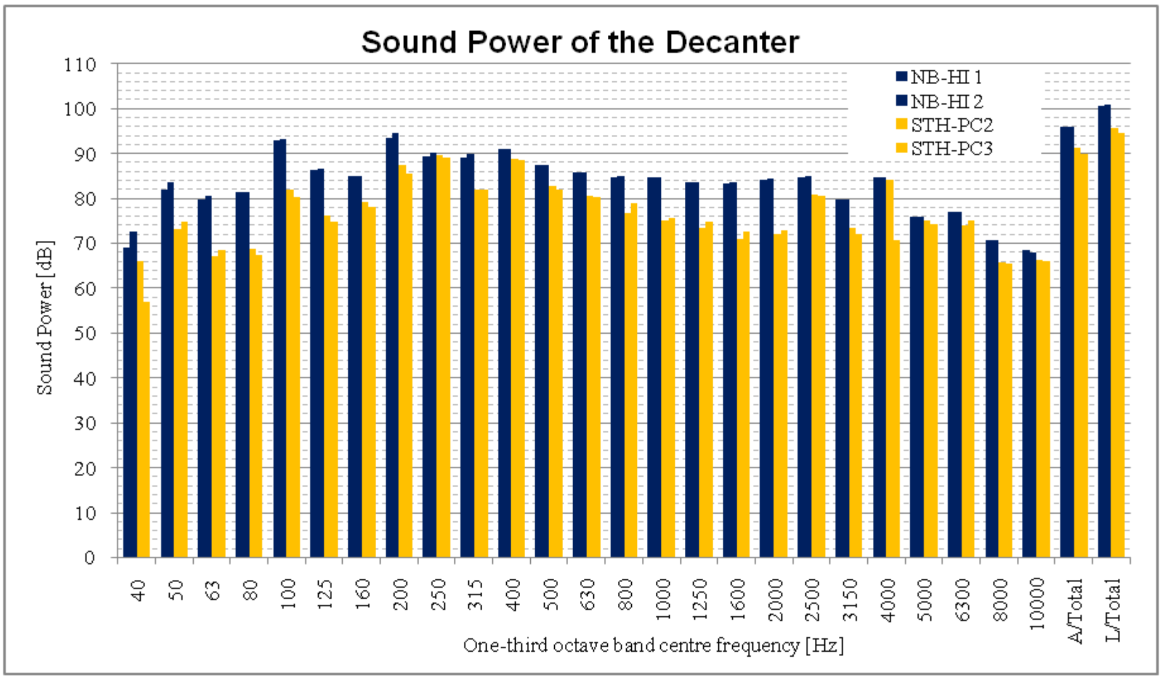


Figure 124 Sound power comparison

Table 28 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-HI1	84.0	86.7	87.1	80.1		90.0	92.8	92.0	85.4	89.3
NB-HI2	85.5	88.0	87.7	79.0		91.5	93.2	92.5	85.8	90.0
STH-PC2	80.7	81.2	82.8	79.5		83.9	86.4	86.1	83.1	83.9
STH-PC3	79.0	82.7	83.7	79.0		82.7	85.1	84.8	82.6	83.3
Ave Diff.	-4.9	-5.4	-4.2	-0.3		-7.4	-7.3	-6.8	-2.8	-6.0
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-HI1	82.9	87.8	87.6	85.3		84.2	92.1	91.7	87.0	88.9
NB-HI2	83.1	88.3	87.9	85.6		84.5	92.6	91.7	87.2	89.2
STH-PC2	79.6	83.7	84.8	83.3		80.7	85.3	85.5	84.2	83.9
STH-PC3	79.9	83.3	83.8	81.0		80.7	84.4	83.8	82.1	82.9
Ave Diff.	-3.3	-4.6	-3.5	-3.3		-3.6	-7.5	-7.1	-3.9	-5.7
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-HI1	86.3	88.1	87.4	89.4	88.6			83.2	86.5	85.3
NB-HI2	86.2	88.3	88.4	90.4	89.4			82.7	86.8	85.4
STH-PC2	80.2	82.8	78.5	83.4	82.5			80.8	83.4	82.4
STH-PC3	78.1	81.8	80.2	83.1	82.1			77.8	82.5	80.9
Ave Diff.	-7.1	-5.9	-8.6	-6.7	-6.7			-3.7	-3.7	-3.7
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-HI1	85.5	86.9	86.8	85.7	86.4					88.4
NB-HI2	86.5	87.3	87.0	85.6	86.7					88.9
STH-PC2	81.1	82.9	82.3	83.5	82.7					83.5
STH-PC3	78.7	82.5	81.2	81.9	81.5					82.6
Ave Diff.	-6.1	-4.4	-5.2	-3.0	-4.5					-5.6

Table 29 Comparison of sound intensity measurements for the 250 Hz one-third octave band [dB]

One-third octave band centre frequency:				250	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-HI1	76.8	76.4	77.8	54.1		80.9	80.6	80.6	75.4	78.4
NB-HI2	78.5	78.2	78.9	0.0		82.1	81.2	80.8	76.8	79.3
STH-PC2	77.9	76.1	79.1	73.2		77.0	78.2	80.5	77.1	78.0
STH-PC3	76.9	77.8	77.9	70.9		77.0	77.8	79.3	77.4	77.5
Ave Diff.	-0.3	-0.4	0.2	18.0		-4.5	-2.9	-0.8	1.2	-1.1
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-HI1	70.2	77.7	77.4	77.2		61.3	81.0	78.7	76.6	77.5
NB-HI2	72.3	78.7	78.5	78.4		68.9	81.8	79.2	76.9	78.4
STH-PC2	75.3	79.6	81.1	80.3		74.2	80.1	78.9	78.5	79.0
STH-PC3	74.3	79.3	79.8	78.1		75.7	79.7	78.5	77.8	78.4
Ave Diff.	3.6	1.2	2.5	1.4		9.9	-1.5	-0.3	1.4	0.7
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-HI1	73.7	74.6	76.9	78.8	77.5			73.4	75.2	74.5
NB-HI2	74.1	74.7	78.9	79.9	78.6			72.5	76.1	74.8
STH-PC2	68.3	73.3	76.7	78.0	76.5			73.7	77.4	76.1
STH-PC3	69.6	73.2	76.1	77.1	75.8			0.0	78.1	75.1
Ave Diff.	-5.0	-1.4	-1.5	-1.8	-1.8			0.8	2.1	1.0
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-HI1	75.4	73.4	75.2	75.2	74.8					77.2
NB-HI2	76.8	74.9	75.6	75.4	75.6					78.1
STH-PC2	73.1	73.9	73.8	75.1	74.1					77.6
STH-PC3	70.8	70.7	72.9	74.0	72.4					76.9
Ave Diff.	-4.2	-1.8	-2.1	-0.8	-1.9					-0.4

Table 30 Comparison of sound intensity measurements for 4,000 Hz one-third octave band [dB]

One-third octave band centre frequency:				4000	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-HI1	71.6	69.7	67.7	65.7		77.0	77.0	72.7	67.7	72.9
NB-HI2	71.0	69.6	67.9	65.2		77.0	76.5	72.6	67.4	72.6
STH-PC2	70.9	66.6	65.1	63.5		76.6	76.0	70.8	66.4	71.7
STH-PC3	56.9	56.9	58.0	59.9		55.0	57.1	57.9	62.4	58.7
Ave Diff.	↓ -7.4	↓ -7.9	↓ -6.3	↓ -3.8		↓ -11.2	↓ -10.2	↓ -8.3	↓ -3.2	↓ -7.6
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-HI1	66.8	67.9	70.0	71.8		68.4	72.9	78.9	78.5	74.2
NB-HI2	66.8	67.6	69.7	71.8		68.9	73.3	79.0	78.3	74.2
STH-PC2	64.6	65.4	68.2	71.0		68.0	72.0	79.0	78.4	73.9
STH-PC3	58.7	55.8	57.1	56.9		62.0	58.0	57.2	55.4	58.3
Ave Diff.	↓ -5.2	↓ -7.2	↓ -7.2	↓ -7.8		↓ -3.7	↓ -8.1	↓ -10.9	↓ -11.5	↓ -8.1
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-HI1	71.3	74.3	71.0	74.7	73.9			65.6	65.8	65.7
NB-HI2	71.0	74.7	70.1	75.2	74.3			64.9	65.6	65.3
STH-PC2	71.1	73.8	71.1	74.5	73.7			63.1	65.0	64.2
STH-PC3	58.0	63.5	57.7	55.3	59.0			58.0	62.0	60.6
Ave Diff.	↓ -6.6	↓ -5.8	↓ -6.1	↓ -10.1	↓ -7.7			↓ -4.7	↓ -2.2	↓ -3.1
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-HI1	69.4	68.0	64.8	64.7	66.9					72.5
NB-HI2	69.9	68.4	65.5	64.7	67.3					72.5
STH-PC2	68.2	65.6	62.6	63.4	65.0					71.8
STH-PC3	56.1	57.5	54.2	59.9	57.4					58.6
Ave Diff.	↓ -7.5	↓ -6.7	↓ -6.8	↓ -3.1	↓ -5.9					↓ -7.3

9.2 Decanter Modification Two

9.2.1 The Modification – Smooth with Ports Open

In the second modification, see Figure 125, the following changes were made:

- The solid discharge ports were opened and taped.
- The third phase liquid discharge ports on the side of the bowl were opened.

a Solid discharge ports opened

b Liquid discharged ports opened

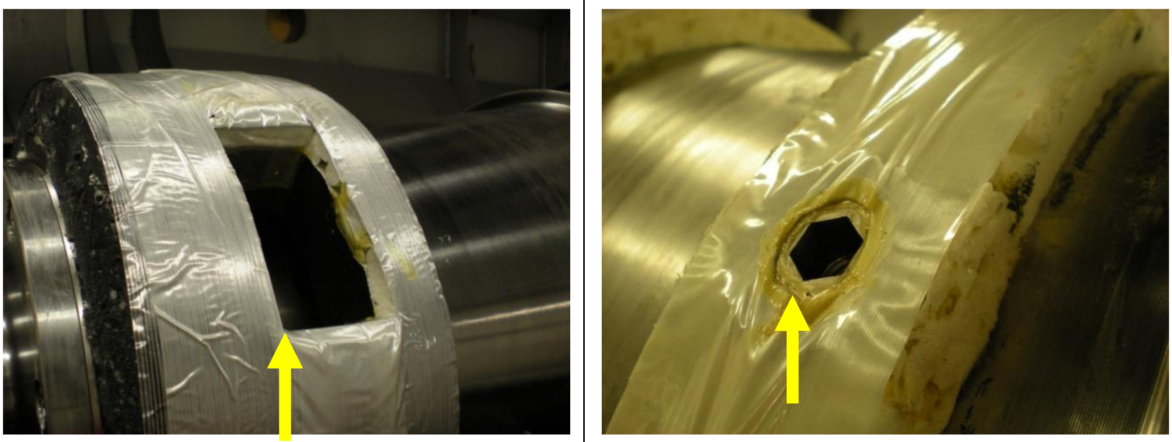


Figure 125 (a-b) Decanter modifications

9.2.2 Results of the Modification

The measured sound powers of the decanter are shown in Figure 126. Except for the first three one-third octave bands, the two scans produced consistent results. There was a requirement to modify the taping used in the modification between the two scans and this could account for the minor variations in the levels recorded. The variations in the three lowest one-third octave bands did not impact the overall results, due to their relatively low sound power levels. Therefore the scans were deemed to be a fair representation of the sound that the decanter produced. The overall sound power of the decanter was 97.3 dB.

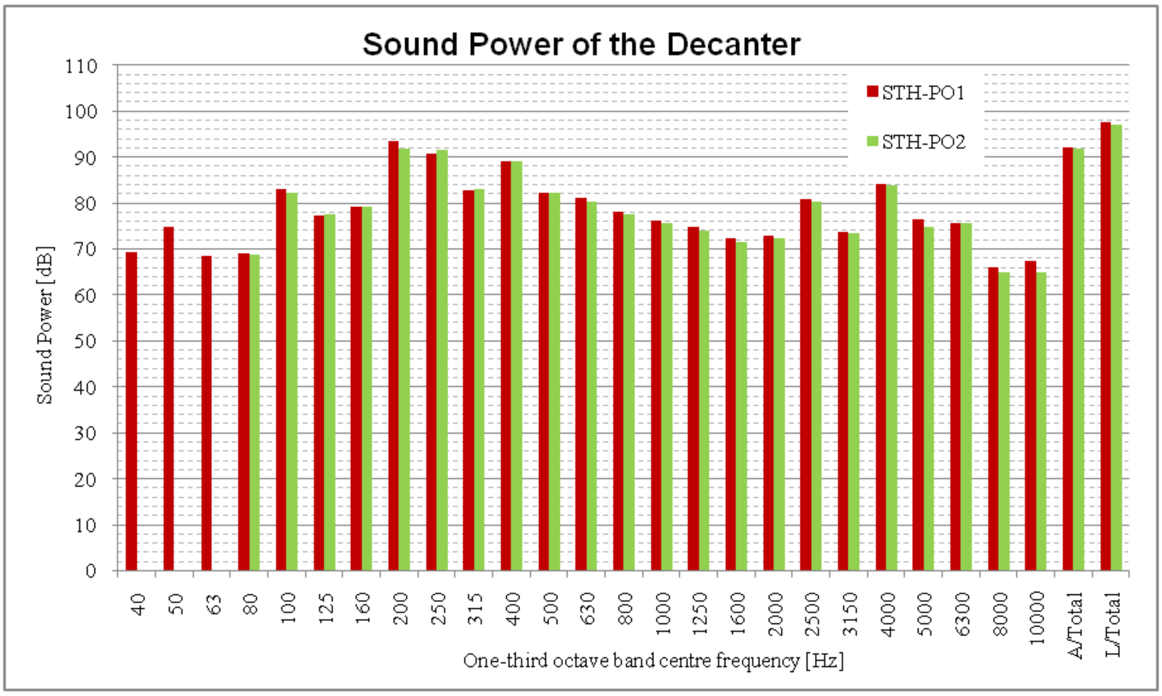


Figure 126 Sound power of the decanter

9.2.3 Comparison with Previous Modification

The sound power of the current decanter configuration, STH-PO, and the previous configuration, STH-PC (see Section 9.1), are compared in Figure 127 and the associated sound intensity levels in Tables 31 and 32. The increase in overall sound power due to opening the port holes was 2.2 dB. A comparison with the NB-HI configuration is also shown in Table 31.

The increase in sound power was mainly due to the 6.1 dB increase in the 200 Hz one-third octave band. Table 32 shows the sound intensity for the 200 Hz one-third octave band. The highest increases, 9 dB and over, in sound intensities were for regions Front 1 and 5, Back 2, 4, 7 and 8, and Top 1. The increases in sound intensity were clustered around the gearbox end of the decanter where there were 4 discharge ports in the end of the bowl. These would combine to produce noise at 216 Hz when the bowl is rotating, which falls within the 200 Hz one-third octave band. It was likely that opening the port holes had enabled the bowl to pump more air through itself and therefore produce more noise through the liquid discharge ports in the hub. This noise would be radiated through the end of the hopper and into regions Front 1 and 5, Back 4 and 8, and Top 1. There are also four solid discharge ports at the other end of the bowl and the regions that correspond to the solid discharge ports, Front 3 and 7 and Back 2 and 6, also had high sound intensity levels. Therefore the increase in sound power was mainly due to the increased air flow through the rotating bowl due to the opening of the port holes.

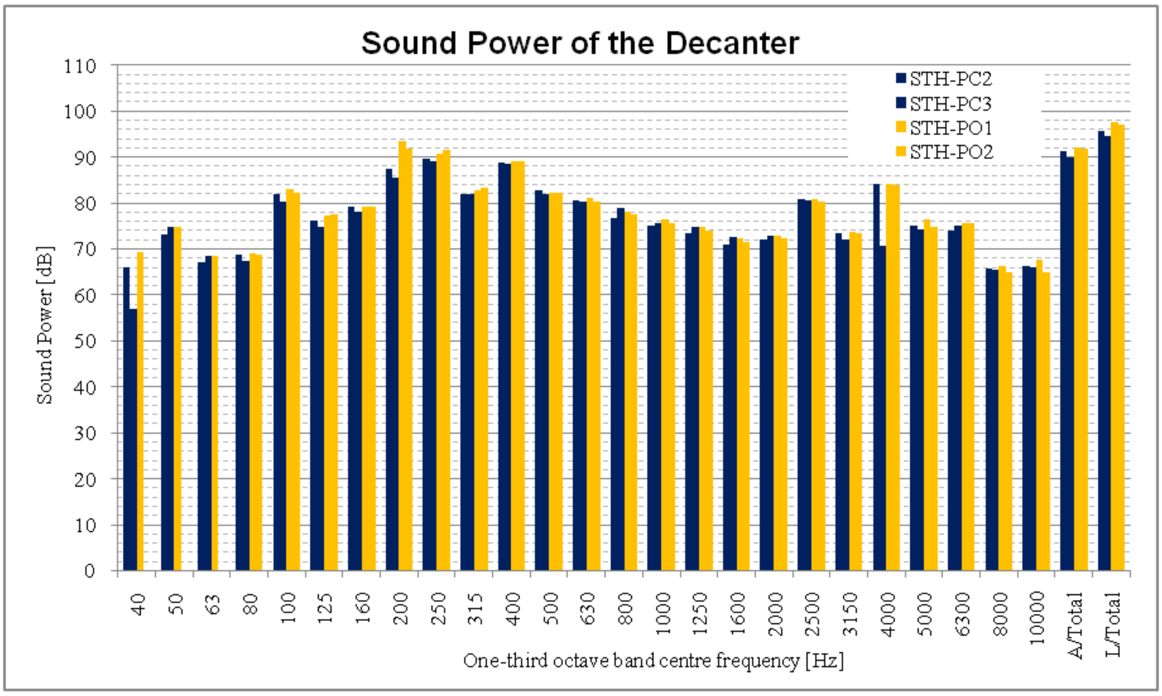


Figure 127 Sound power comparison

Table 31 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:				L/Total	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
STH-PC2	80.7	81.2	82.8	79.5		83.9	86.4	86.1	83.1	83.9
STH-PC3	79.0	82.7	83.7	79.0		82.7	85.1	84.8	82.6	83.3
STH-PO1	82.4	84.6	84.9	76.7		88.4	88.3	87.8	84.1	86.0
STH-PO2	84.3	84.9	84.5	78.9		86.9	87.5	87.1	84.5	85.6
Ave Diff.	⇧ 3.5	⇧ 2.8	⇧ 1.5	⇩ -1.4		⇧ 4.3	⇧ 2.2	⇧ 2.0	⇧ 1.5	⇧ 2.2
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
STH-PC2	79.6	83.7	84.8	83.3		80.7	85.3	85.5	84.2	83.9
STH-PC3	79.9	83.3	83.8	81.0		80.7	84.4	83.8	82.1	82.9
STH-PO1	80.0	86.5	85.3	83.9		80.1	89.8	87.2	86.9	86.3
STH-PO2	80.5	86.0	85.4	84.2		79.5	88.1	87.2	86.5	85.7
Ave Diff.	⇨ 0.5	⇧ 2.8	⇧ 1.1	⇧ 1.9		⇨ -0.9	⇧ 4.1	⇧ 2.6	⇧ 3.6	⇧ 2.6
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
STH-PC2	80.2	82.8	78.5	83.4	82.5			80.8	83.4	82.4
STH-PC3	78.1	81.8	80.2	83.1	82.1			77.8	82.5	80.9
STH-PO1	80.3	82.2	83.5	86.6	85.1			79.7	84.6	83.0
STH-PO2	80.0	82.5	83.0	85.5	84.2			80.6	84.5	83.1
Ave Diff.	⇨ 1.0	⇨ 0.0	⇧ 3.9	⇧ 2.8	⇧ 2.3			⇨ 0.9	⇧ 1.6	⇧ 1.4
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
STH-PC2	81.1	82.9	82.3	83.5	82.7					83.5
STH-PC3	78.7	82.5	81.2	81.9	81.5					82.6
STH-PO1	82.5	84.0	83.8	83.9	83.7					85.4
STH-PO2	82.7	84.1	83.9	84.0	83.8					85.0
Ave Diff.	⇧ 2.7	⇧ 1.3	⇧ 2.1	⇧ 1.3	⇧ 1.7					⇧ 2.2

Table 32 Comparison of sound intensity measurements for the 200 Hz one-third octave band [dB]

One-third octave band centre frequency:				200	Hz					
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
STH-PC2	69.5	74.7	75.8	73.6		77.2	78.7	80.0	76.9	77.1
STH-PC3	69.9	73.8	74.9	71.6		76.3	77.0	78.4	75.5	75.7
STH-PO1	80.1	80.7	81.1	0.0		86.5	84.8	83.8	80.0	82.5
STH-PO2	82.6	81.9	78.9	0.0		85.1	82.9	81.0	78.2	81.2
Ave Diff.	⇧ 11.7	⇧ 7.1	⇧ 4.7	-		⇧ 9.1	⇧ 6.0	⇧ 3.2	⇧ 2.9	⇧ 5.4
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
STH-PC2	64.7	74.6	77.0	68.5		0.0	78.1	74.1	75.2	74.6
STH-PC3	66.1	66.7	72.7	66.9		63.6	77.3	68.4	71.4	71.8
STH-PO1	0.0	82.0	79.8	77.8		0.0	87.2	81.9	83.8	82.1
STH-PO2	0.0	79.7	78.9	77.3		0.0	84.6	80.8	81.2	80.0
Ave Diff.	-	⇧ 10.2	⇧ 4.5	⇧ 9.8		-	⇧ 8.2	⇧ 10.1	⇧ 9.2	⇧ 7.9
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
STH-PC2	65.1	0.0	73.6	76.5	74.1			70.7	72.7	71.9
STH-PC3	67.1	63.5	71.9	74.6	72.7			67.1	70.5	69.2
STH-PO1	0.0	0.0	80.5	84.0	81.3			67.7	78.6	76.2
STH-PO2	0.0	0.0	79.5	82.7	80.1			71.6	76.9	75.2
Ave Diff.	-	-	⇧ 7.3	⇧ 7.8	⇧ 7.3			⇨ 0.8	⇧ 6.2	⇧ 5.1
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
STH-PC2	68.9	74.6	75.1	75.1	74.4					75.3
STH-PC3	64.5	71.1	73.2	74.7	72.5					73.5
STH-PO1	76.4	77.1	78.4	77.5	77.5					81.2
STH-PO2	76.8	76.6	78.6	77.3	77.5					79.8
Ave Diff.	⇧ 9.9	⇧ 4.0	⇧ 4.3	⇧ 2.5	⇧ 4.1					⇧ 6.1

Table 33 Comparison of sound intensity measurements for 4,000 Hz one-third octave band [dB]

One-third octave band centre frequency:				4000 Hz						
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
NB-HI1	71.6	69.7	67.7	65.7		77.0	77.0	72.7	67.7	72.9
NB-HI2	71.0	69.6	67.9	65.2		77.0	76.5	72.6	67.4	72.6
STH-PO1	70.9	67.5	65.6	64.9		76.5	75.8	70.9	66.7	71.7
STH-PO2	70.0	66.6	65.2	64.1		76.9	75.8	71.1	67.1	71.8
Ave Diff.	⇨ -0.8	⇩ -2.6	⇩ -2.4	⇨ -1.0		⇨ -0.3	⇨ -1.0	⇩ -1.7	⇨ -0.7	⇩ -1.0
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
NB-HI1	66.8	67.9	70.0	71.8		68.4	72.9	78.9	78.5	74.2
NB-HI2	66.8	67.6	69.7	71.8		68.9	73.3	79.0	78.3	74.2
STH-PO1	64.9	65.4	68.2	71.1		67.9	71.9	79.0	78.8	74.0
STH-PO2	64.0	64.1	66.0	69.4		66.2	71.8	78.9	78.5	73.6
Ave Diff.	⇩ -2.3	⇩ -3.0	⇩ -2.8	⇩ -1.6		⇩ -1.6	⇩ -1.3	⇨ 0.0	⇨ 0.3	⇨ -0.4
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
NB-HI1	71.3	74.3	71.0	74.7	73.9			65.6	65.8	65.7
NB-HI2	71.0	74.7	70.1	75.2	74.3			64.9	65.6	65.3
STH-PO1	71.2	74.4	71.1	74.8	74.0			63.4	65.4	64.6
STH-PO2	69.3	73.0	70.1	73.5	72.7			63.3	64.5	64.0
Ave Diff.	⇨ -0.9	⇨ -0.8	⇨ 0.0	⇨ -0.8	⇨ -0.8			⇩ -1.9	⇨ -0.7	⇩ -1.2
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
NB-HI1	69.4	68.0	64.8	64.7	66.9					72.5
NB-HI2	69.9	68.4	65.5	64.7	67.3					72.5
STH-PO1	69.4	66.2	62.9	64.1	65.8					72.0
STH-PO2	69.8	66.1	62.7	63.4	65.8					71.6
Ave Diff.	⇨ -0.1	⇩ -2.1	⇩ -2.4	⇨ -1.0	⇩ -1.3					⇨ -0.7

9.3 Decanter Modification Three

9.3.1 The Modification – Hopper Holes Open

The third modification was the same as the previous modification but with the plates covering the hopper holes removed, see Figures 121 (d) and 122 (e).

9.3.2 Results of the Modification

The sound power of the decanter is shown in Figure 128. Except for the first two one-third octave bands, the two scans produced consistent results, even though there was a requirement to modify the taping used in the modification between the two scans. The sound power of this configuration was 98.8 dB.

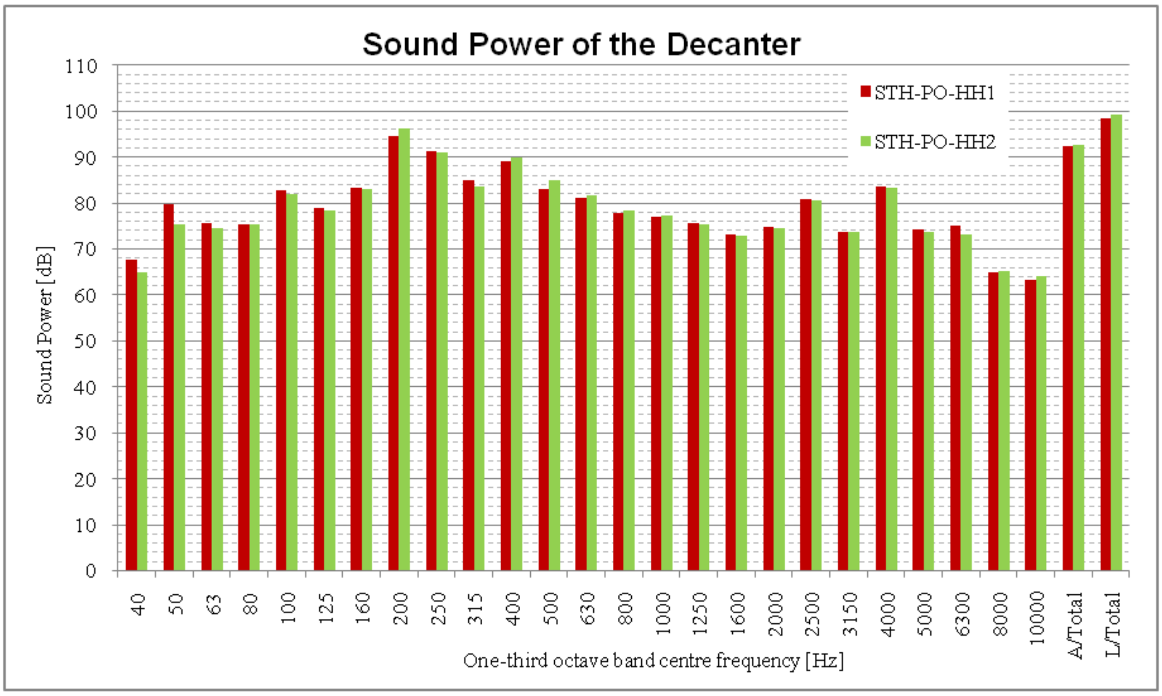


Figure 128 Sound power of the decanter

9.3.3 Comparison with Previous Configurations

The comparison of the sound power of the current decanter configuration, STH-PO-HH, and the previous configuration, STH-PO (see Section 9.2), are shown in Figure 129 and the sound intensity values are compared in Table 34. The effect of removing the covers from the hopper holes was an increase in sound power of 1.4 dB. The largest increases were in the 80 Hz, 160 Hz and 200 Hz one-third octave bands. The current configuration had a sound power 1.4 dB higher than that for the previous configuration.

The increase in sound power was mainly due to a 2.8 dB increase in the 200 Hz one-third octave band. Table 34 shows the sound intensity for the 200 Hz one-third octave band. The highest increases, 4 dB and over, in sound intensities were for regions Front 6, Back 3, 4 and 5, and Top 1. These regions do not directly relate to the holes in the hopper, which are Front 7 and Back 6. The increases in sound intensity are clustered around the gearbox end of the bowl where the liquid discharge ports are. There were 4 discharge ports in the end of the bowl and these would combine to produce noise at 216 Hz when the bowl is rotating, which is within the 200 Hz one-third octave band. It was likely that opening the hopper holes enabled the bowl to pump more air through itself and therefore produce more noise through the liquid discharge ports in the hub. Therefore the increase in sound power was mainly due to the increased flow through the rotating bowl due to opening of the hopper holes.

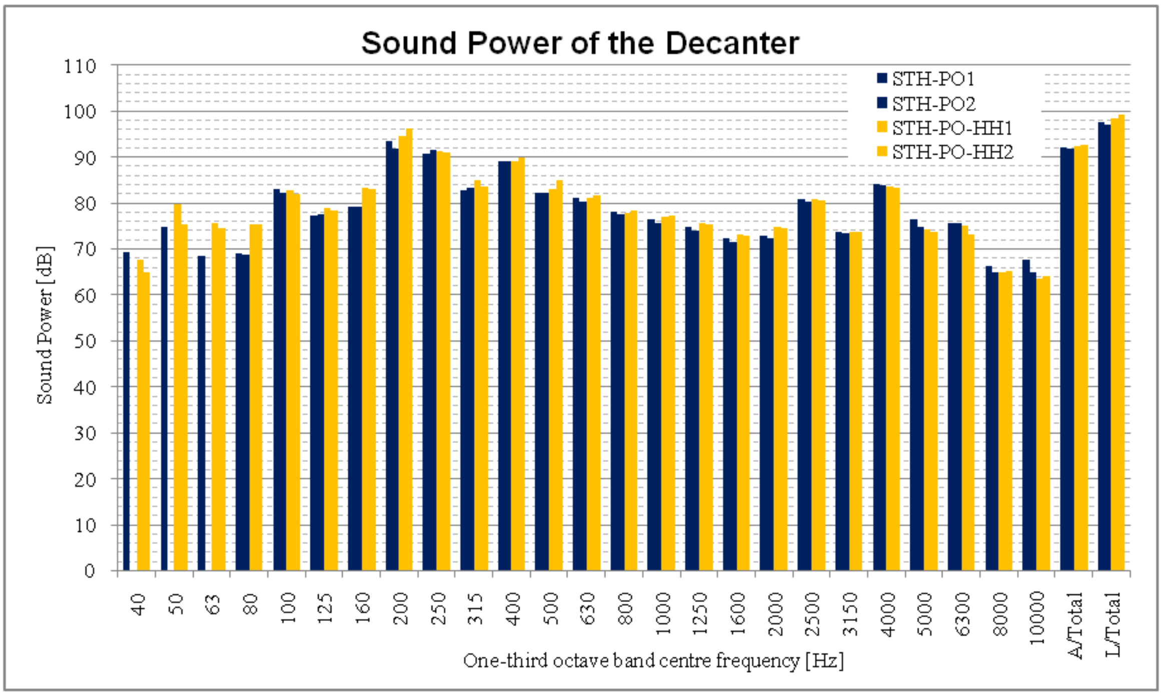


Figure 129 Sound power comparison

Table 34 Comparison of overall sound intensity [dB]

One-third octave band centre frequency:			200	Hz						
Scan	Front 1	Front 2	Front 3	Front 4		Front 5	Front 6	Front 7	Front 8	Front - Intensity
STH-PO1	80.1	80.7	81.1	0.0		86.5	84.8	83.8	80.0	82.5
STH-PO2	82.6	81.9	78.9	0.0		85.1	82.9	81.0	78.2	81.2
STH-PO-HH1	76.6	81.2	78.8	0.0		88.0	87.3	85.9	80.0	83.8
STH-PO-HH2	81.2	83.5	81.1	0.0		89.8	89.6	86.6	79.3	85.6
Ave Diff.	↓ -2.4	↑ 1.0	⇨ -0.1	-		↑ 3.1	↑ 4.6	↑ 3.8	⇨ 0.6	↑ 2.9
Scan	Back 1	Back 2	Back 3	Back 4		Back 5	Back 6	Back 7	Back 8	Back - Intensity
STH-PO1	0.0	82.0	79.8	77.8		0.0	87.2	81.9	83.8	82.1
STH-PO2	0.0	79.7	78.9	77.3		0.0	84.6	80.8	81.2	80.0
STH-PO-HH1	0.0	82.3	82.1	81.1		0.0	86.9	84.2	84.0	82.8
STH-PO-HH2	0.0	84.3	84.5	84.0		0.0	88.5	86.6	84.7	84.7
Ave Diff.	-	↑ 2.5	↑ 4.0	↑ 5.0		-	↑ 1.8	↑ 4.1	↑ 1.8	↑ 2.7
Scan	Left 1	Left 2	Left 3	Left 4	Left - Intensity			Right 1	Right 2	Right - Intensity
STH-PO1	0.0	0.0	80.5	84.0	81.3			67.7	78.6	76.2
STH-PO2	0.0	0.0	79.5	82.7	80.1			71.6	76.9	75.2
STH-PO-HH1	68.9	0.0	82.8	84.9	82.6			69.1	78.4	76.1
STH-PO-HH2	71.3	0.0	83.7	86.1	83.6			69.0	81.5	79.0
Ave Diff.	-	-	↑ 3.3	↑ 2.2	↑ 2.4			⇨ -0.6	↑ 2.2	↑ 1.9
Scan	Top 1	Top 2	Top 3	Top 4	Top - Intensity					Total - Intensity
STH-PO1	76.4	77.1	78.4	77.5	77.5					81.2
STH-PO2	76.8	76.6	78.6	77.3	77.5					79.8
STH-PO-HH1	83.1	79.9	79.1	79.1	80.2					82.4
STH-PO-HH2	83.7	80.4	78.7	78.6	80.4					84.0
Ave Diff.	↑ 6.8	↑ 3.3	⇨ 0.4	↑ 1.4	↑ 2.8					↑ 2.7

9.4 Discussion

9.4.1 Smoothing the Bowl

Smoothing of the bowl was undertaken to determine the level of sound power generated by turbulence from the rotating bowl. The bowl was smoothed by the use of expandable foam, tape, glue and silicon. The holes in the hopper were blocked with the use of 3 mm steel plate and silicon. The bowl was not perfectly smooth; the photos in Figures 122 and 125 show that the tape and foam did not produce a perfectly smooth surface. This indicates that the results achieved could probably be improved upon by achieving a smoother finish.

The configuration with smoothed bowl and closed discharge ports had a sound power of 95.1 dB. This was a 5.6 dB reduction from the previous configuration of new base-hopper isolated. This represents more than 70 % reduction in sound power. For the one-third octave bands that contained the rotating bowl harmonics the reductions in sound power were between 80 % and 94 %. The results showed that the turbulence from the bowl was the most significant contributor to the sound power of the decanter.

9.4.2 Smoothing the Solid Discharge Ports

The smoothing of the solid discharge ports reduced turbulence in the following ways:

- Turbulence from the four port holes relates to reductions in the 200 Hz one-third octave band (as well as reducing the air flow through the four liquid discharge ports in the other end of the bowl).
- Turbulence from the two wiper ploughs relates to reductions in the 100 Hz one-third octave band.
- Turbulence from the gaps between the wear plates.

Figure 130 shows the design of the solid discharge ports in grey with a possible redesign of the solid discharge ports shown to the right. The current design has many edges from which turbulence is generated. The new concept maintains the two wiper ploughs but has removed all the trailing edges that generate turbulence. The possible redesign is idealised as removable rear plates need to be incorporated, but these need to be set in flush with trailing edges minimised.

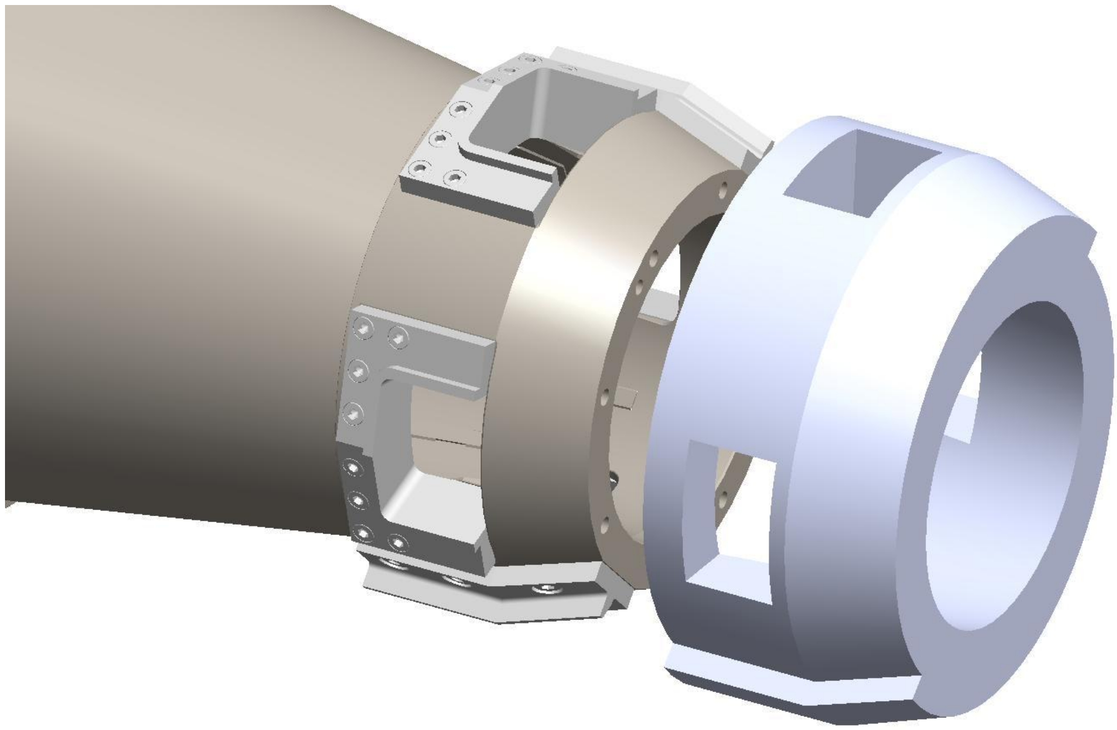


Figure 130 Redesign of the solid discharge port

9.4.3 Smoothing the Third Phase Liquid Discharge Ports

The smoothing of the third phase liquid discharge ports reduced turbulence in the 315 Hz one-third octave band. Figure 121 (a) shows the design of the third phase liquid discharge ports on the side of the bowl where the protrusions were welded onto the bowl. The suggested redesign of the ports is shown in Figure 131. The redesign of the ports has the added benefit of less machining of the bowl and the removal of possible fatigue cracking due to stress corrosion.

The smoothing of the bolt heads and bolt holes reduced turbulence in the 800 Hz and 1,600 Hz one-third octave bands. The bolt holes were drilled right through the bowl flanges, as shown in Figure 121 (b). These holes could be left blind and ensure a smooth finish. However, the tapping of the holes would be more complicated requiring depth control and cleanout.

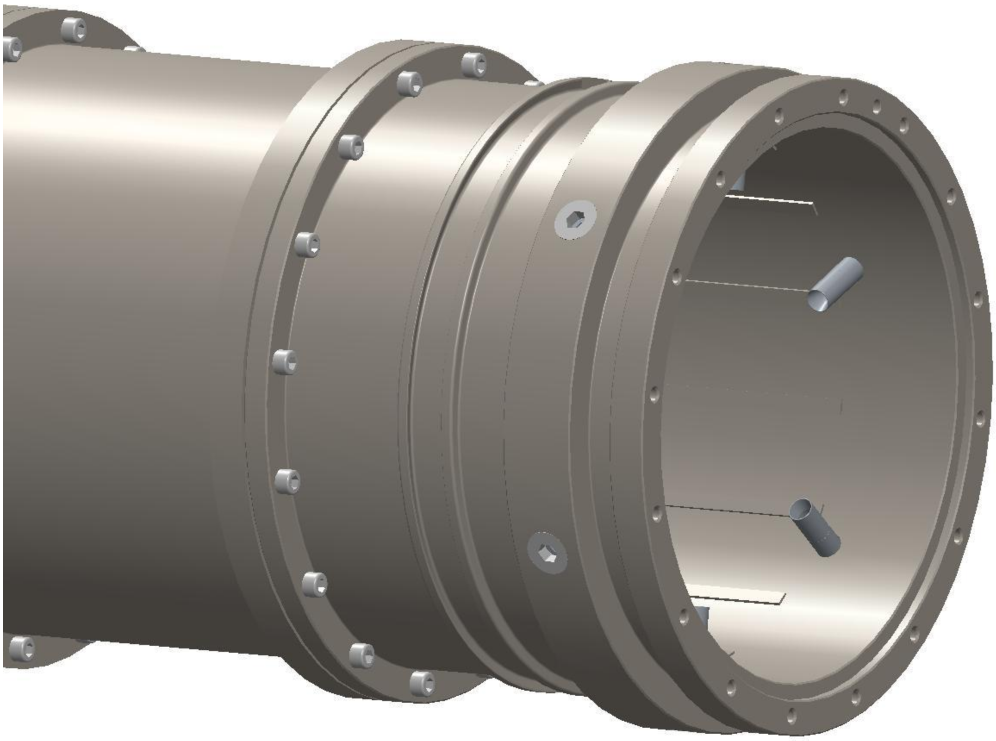


Figure 131 Redesign of the third phase liquid discharge ports on the side of the bowl

9.4.4 Smoothing the Bolt Heads and Balance Ring

There are two ways to reduce turbulence due to the bolt heads. The bowl flange could be made thicker and the bolt heads countersunk. This may not be a practical option due to the hole's close proximity to the bowl surface. A second option could be to use a cover over the bolt heads as shown in Figure 132. The cover could be held in place by several of the 16 bolts that are used to hold the bowl together. This option could be also retrofitted to existing decanters.

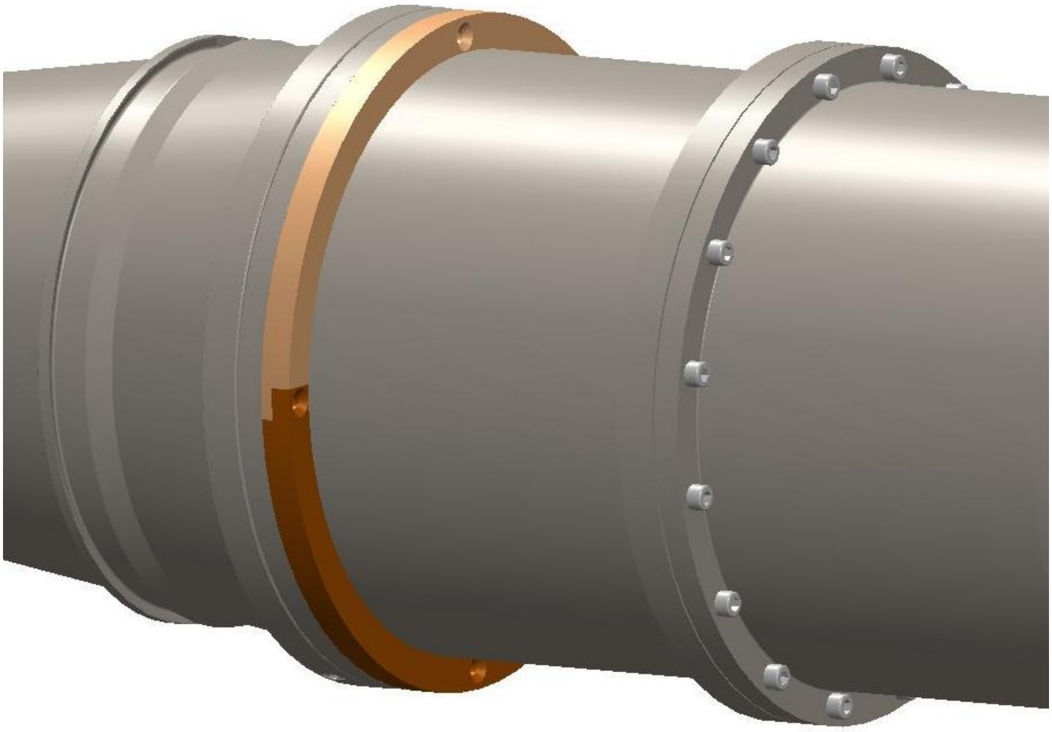


Figure 132 Redesign of the bolted flange

The turbulence from the balancing ring, see Figure 121 (b), could be reduced by using a sleeve that slides over the entire balancing ring. The sleeve would be from of rubber with wire beads, similar to a tyre, and held in position by the outer flanges of the balancing ring.

9.4.5 Reducing Noise from the Liquid Discharge Ports

When the discharge ports were opened the sound power increased by 2.2 dB. This increase in sound power was primarily due to a 6.1 dB increase in the 200 Hz one-third octave band. The increase in the 200 Hz one-third octave band was attributed to an increase in air flow through the bowl and an associated increase in noise from the four liquid discharge ports in the liquid end hub.

When the hopper holes were opened there was a 1.4 dB increase in sound power. This increase was primarily due to a 2.8 dB increase in the 200 Hz one-third octave band. The increase in the 200 Hz one-third octave band was again attributed to an increase in air flow through the bowl and an associated increase in noise from the four liquid discharge ports in the liquid end hub.

The 200 Hz one-third octave band is the main source of sound power for the smoothed bowl and ports open. The main source for noise for the 200 Hz one-third octave band was due to air flow through the bowls solid discharge ports and the liquid discharge ports on the end of the bowl. Reducing the noise from the solids discharge port has been discussed above. Figure 121 (a) shows the liquid discharge ports in the end of the bowl. Reducing noise from the liquid discharge ports on the end of the bowl can be achieved by four options:

- The exit ports are not flush with the end of the bowl. This means there are additional edges where turbulence can be generated. The exit ports need to be brought flush with the end of the hub and the bolt heads countersunk to produce a smooth surface.
- The number of exit ports could be reduced from four to three. This would move the noise produced down to the 160 Hz one-third octave band which had a sound power 13.5 dB lower than the 200 Hz one-third octave band. This may require a redesign of the shape and size of the port holes to ensure the through put and pond depth of the decanter is not affected.
- The use of a Helmholtz resonator to absorb 216 Hz from within the hopper cavity.

If the number of exit ports were reduced from four to three, the shape of the exit ports would need to change from a circular shape to elliptical to ensure the same flow rate through the exit ports. The area of the four circular holes is $A = \pi D^2$ and three ellipse holes is $A = \frac{3}{4}\pi AB$, see Figure 133. If B = D then $A = \frac{4}{3}$. An ellipse is a circle that has been stretched. If the circle and ellipse are divided by a horizontal line (red dotted line) then the area below the line will be the same for the four circles as the three ellipses. As the pool of liquid in the bowl is evenly spread around the bowl it will form a curved profile at the exit ports (see blue dotted line). This would mean that three elliptical holes would allow more flow compared to four circles. In order for the three elliptical holes to have the same flow rate as four circular holes, the ellipses need to be generated from circles stretched along arks based on the centre of the bowl. Generating the three 'elliptical' holes in this manner would result in the same flow rate, irrespective of pool depth, as the four circular holes.

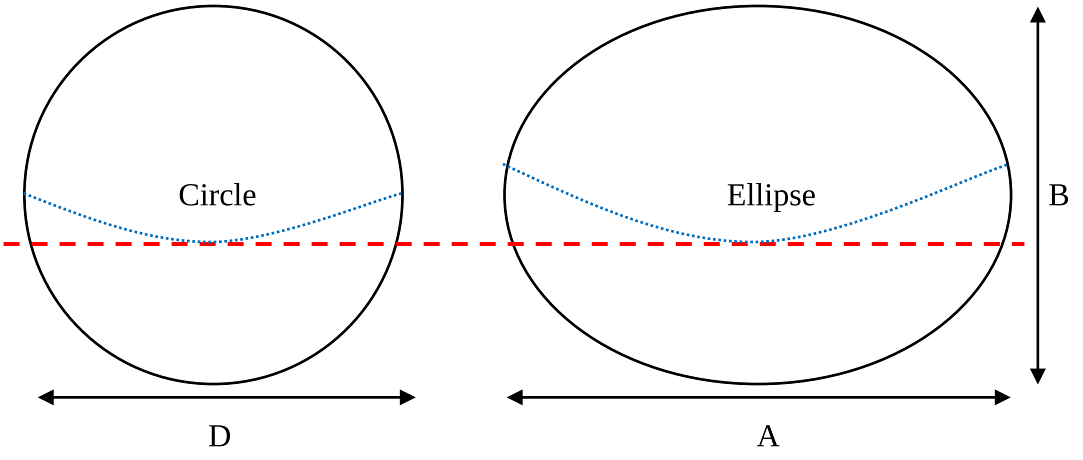


Figure 133 Circle and ellipse area

If the above actions are implemented and able to reduce the sound power level of the 200 Hz and 400 Hz one-third octave bands down below 86 dB, then the overall sound power will reduce down to 95.5 dB.

There was no reduction in sound power for the 250 Hz and 4,000 Hz one-third octave bands. This indicates that the turbulence from the rotating bowl was not the main source of sound power for these two one-third octave bands.

9.5 Conclusion

Turbulence from the rotating bowl has been shown to be a significant contributor to the sound power of the decanter. Smoothing the bowl produced significant sound power reductions. The following actions are recommended in order to reduce the noise due to turbulence of the rotating bowl:

- The number of liquid discharge ports on the end of the bowl is reduced from four to three.
- The liquid discharge ports be redesigned to ensure the decanter's throughput and pond depth is maintained.
- The liquid discharge ports on the end of the bowl are made flush with the end of the bowl.
- The gaps between the third phase liquid discharge ports on the side of the bowl should be filled in, as shown in Figure 131.

- The solid discharge ports should be redesigned to reduce edges that generate turbulence, as shown in Figure 130.
- A cover should be put over the bolt heads, as shown Figure 132.
- The bolt holes should not be drill right through the bowl flanges.
- A sleeve, similar to a tyre, should be placed over the balancing ring.

These actions to reduce turbulence noise could reduce the sound power of the decanter to 95.5 dB.

10 Conclusion

10.1 What the Measurements Showed

The decanter has undergone a number of modifications and the sound power of the decanter resulting from each modification determined. The original configuration was as produced from the factory. The first change was isolating the gearbox guard from the base (OC-GI). The second modification was changing the base from cast iron to polymer concrete (New Base). The measurement results of these three decanter configurations are shown in Figure 134.

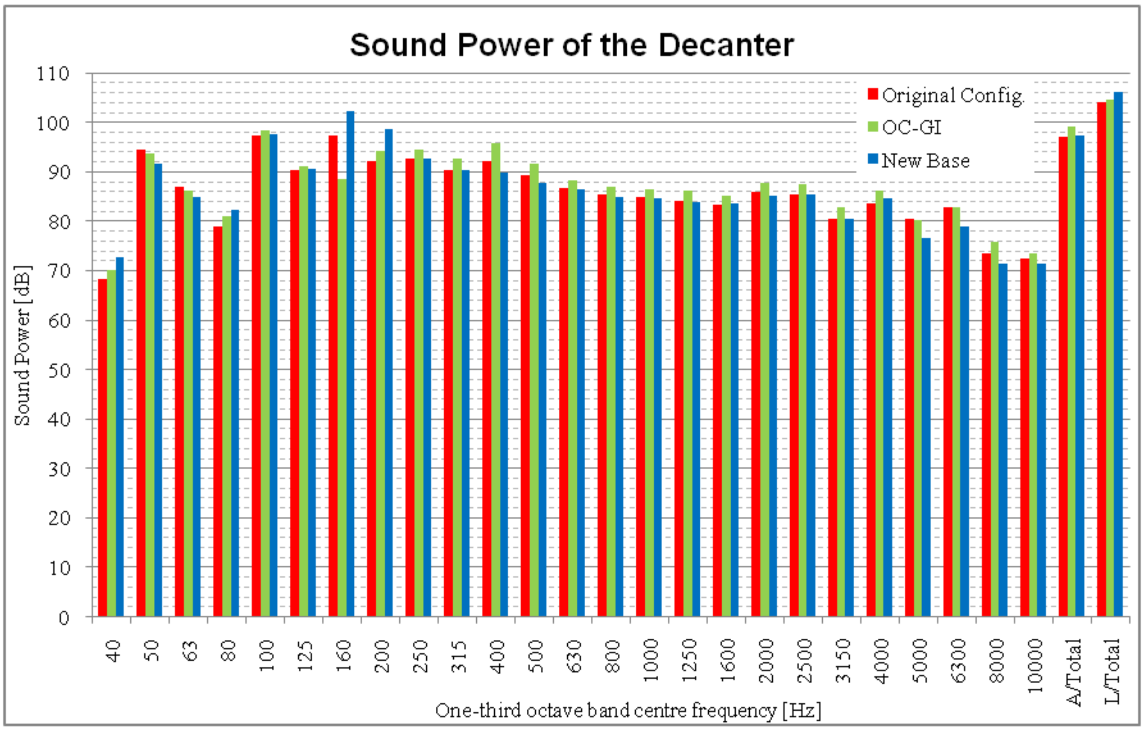


Figure 134 Decanter sound power

The reason for isolating the gearbox guard was due to high vibrations of the gearbox guard at 162 Hz with the 160 Hz one-third octave band consequently having the highest sound power level in the original configuration. The effect of the modification was a significant reduction in sound power from the gearbox guard in the 160 Hz one-third octave band. But the overall sound power had a 0.7 dB increase in sound power due to increases in vibration levels in the base and hopper.

The reason to change the base from a cast iron to polymer concrete was to increase the amount of damping within the base. The result of the modification was a 2 dB increase in sound power when compared to the original configuration. This was primarily due to increases in the 160 Hz and 200 Hz one-third octave bands sound power levels. The increase in these two one-third octave bands correlated to increases in vibrational level within the hopper and gearbox guard at 162 Hz and 216 Hz.

Based on the New Base configuration, three more modifications were evaluated. The first was isolation of the gearbox guard, NB-GI. The second was isolation of the hopper (with gearbox guard isolated), NB-HI. The third modification was the addition of tuned mass dampers to the base (with hopper and gearbox guard isolated), NB-TMD. The sound power measurements are shown in Figure 135.

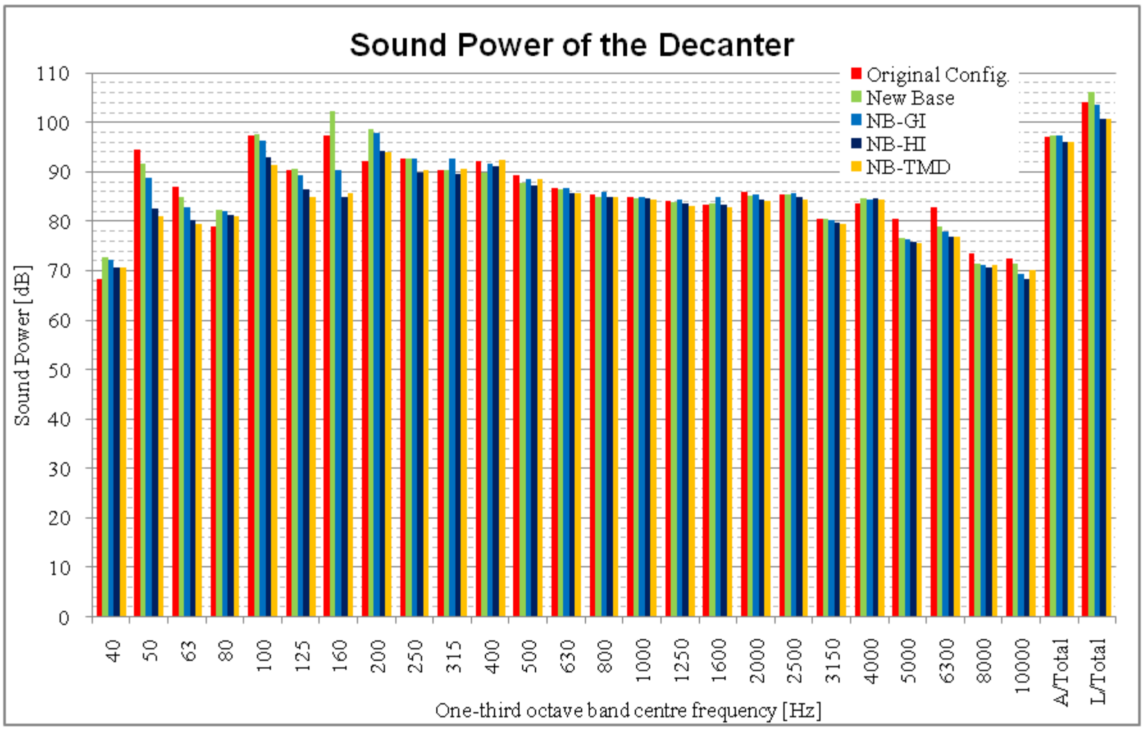


Figure 135 Decanter sound power

The reason for isolating the gearbox guard was due to high vibrations in the gearbox guard at 162 Hz with consequently the 160 Hz one-third octave band having the highest sound power level in the original configuration. The effect of the modification was a significant reduction in sound power from the gearbox guard and the 160 Hz one-third octave band. Isolation of the gearbox guard resulted in a 2.6 dB decrease in sound power.

The isolation of the hopper was undertaken for similar reasons as isolating the gearbox guard. The modification resulted in another 2.7 dB decrease in sound power. The reduction in sound power was due to decreases in the 250 Hz one-third octave band and below.

The reason for the addition of tuned mass dampers was to reduce the vibration levels within the base and hence reduce the sound power level of the decanter. The tuned mass dampers did result in reductions in vibration levels at the frequencies of the tuned mass dampers but did not result in a change in the decanter's sound power level, due to an increase in the 400 Hz one-third octave band.

The next set of modifications were based on reducing the turbulence noise. The modifications were based on the NB-HI configuration of the decanter. The first modification, STH-PC, was to smooth the bowl (including blocking the solid and third phase discharge ports) and blocking off the openings in lower surface of the hopper. The second modification, STH-PO, was the same as the first with the solid and third phase discharge ports opened. The third modification, STH-PO-HH was the same as the second with the openings in lower surface of the hopper opened. The measurement results of these three decanter configurations are shown in Figure 136.

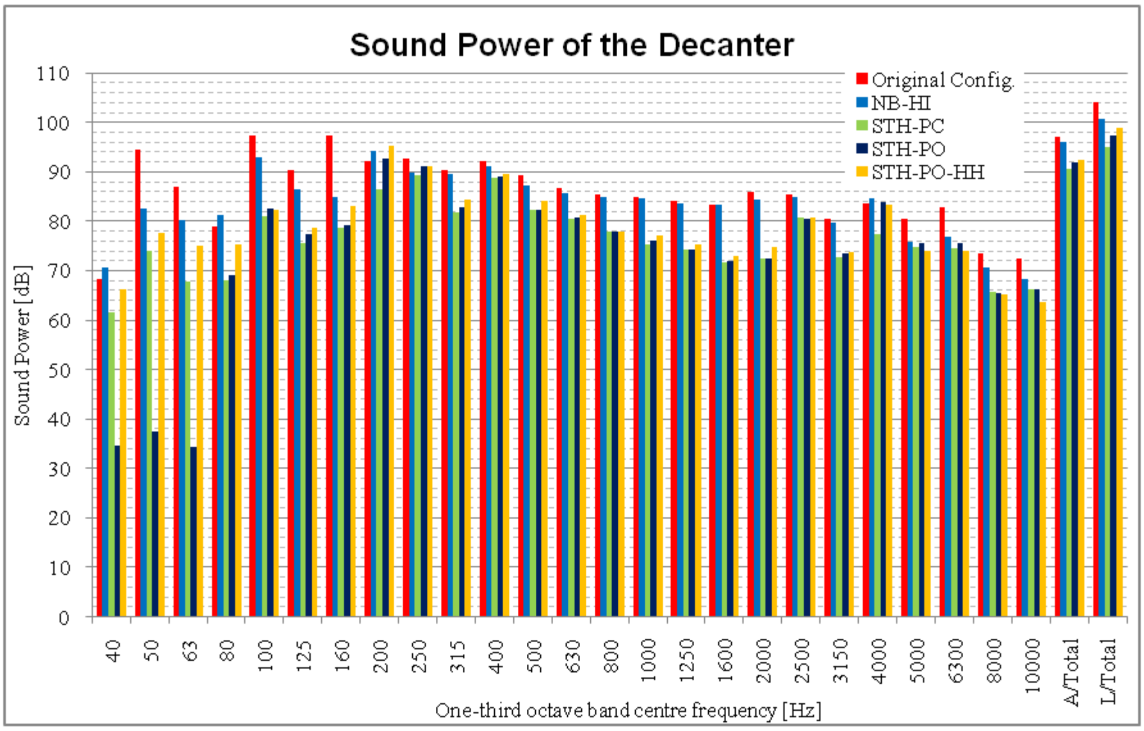


Figure 136 Decanter sound power

The STH-PC modification resulted in reduction in sound power levels across all frequencies and the overall sound power level decreased by 5.6 dB. This was a 75% reduction in sound power. The results indicate that turbulence noise from the rotating bowl was the predominant source of sound power for the decanter in the NB-HI configuration.

The STH-PO modification resulted in a reduction in sound power level of 3.4 dB when compared to the NB-HI configuration and an increase in sound power of 2.2 dB when compared to the STH-PC configuration. The results indicate that the noise due to air flow through the rotating bowl was a significant contributor to the overall sound power of the decanter. The STH-PO-HH modification resulted in an increase in sound power of 1.5 dB when compared to the STH-PO configuration. The results indicate that the hopper has an important role as a sound enclosure.

10.2 Recommendations

From the work performed it is recommended that the decanter is modified to achieve the STH-PO configuration. This requires the following steps to be implemented:

- Isolation of the gearbox guard and hopper from the base. The base requires additional damping, such as using a polymer concrete base, for the isolation of the two components to be effective.
- Smoothing of the bowl surface so that vortices from edges are minimised.
- Removing the holes within the hopper.

It is also recommended that the number of liquid discharge ports in the end of the bowl is reduced from four to three. This would decrease the sound power level of the 200 Hz one-third octave band which was the one-third octave band with the highest sound power level.

10.3 Additional Areas of Research

By changing the decanter from the original configuration to the STH-PO configuration a 7.8 dB reduction in sound power has shown to be achievable. This reduction in sound power was achieved by reducing vibrational/structural noise and vortex/turbulence noise from the rotating bowl. Two main areas need further investigation, the transmission loss through the hopper and the source of the 250 Hz one-third octave band noise.

To further reduce the noise due to vortex/turbulence, the transmission loss through the hopper should be investigated. The hopper was not completely sealed as mechanical labyrinth seals are used at each end of the hopper. The mechanical labyrinth seals are adjacent to the solid and liquid discharge ports which have been identified as significant contributors to the overall sound power. As the mechanical labyrinth seals provide a direct sound path to the surrounding environment, the use of lip seals should be considered as a means of sealing the hopper and lowering the overall sound power of the decanter.

The hopper was also constructed with a series of partitions, which can be seen in Figure 121 (A), which means that the hopper was essentially a series of small panels. Due to the size of the panels the resonant frequencies of the panels needs to be investigated to ensure that they do not coincide with the rotating bowl's harmonics and hence provide minimal transmission loss at those frequencies. The design of the hopper and the materials used in its construction are also areas where an increases in transmission loss can be achieved and therefore reductions in overall sound power for the decanter.

The reduction in noise generated by vibrational/structural noise and vortex/turbulence noise mechanisms means that other mechanisms are now more prominent. Another area of investigation should be focused on the source for the 250 Hz one-third octave band. Likely areas for investigation are the electric motors and their belt drive systems.

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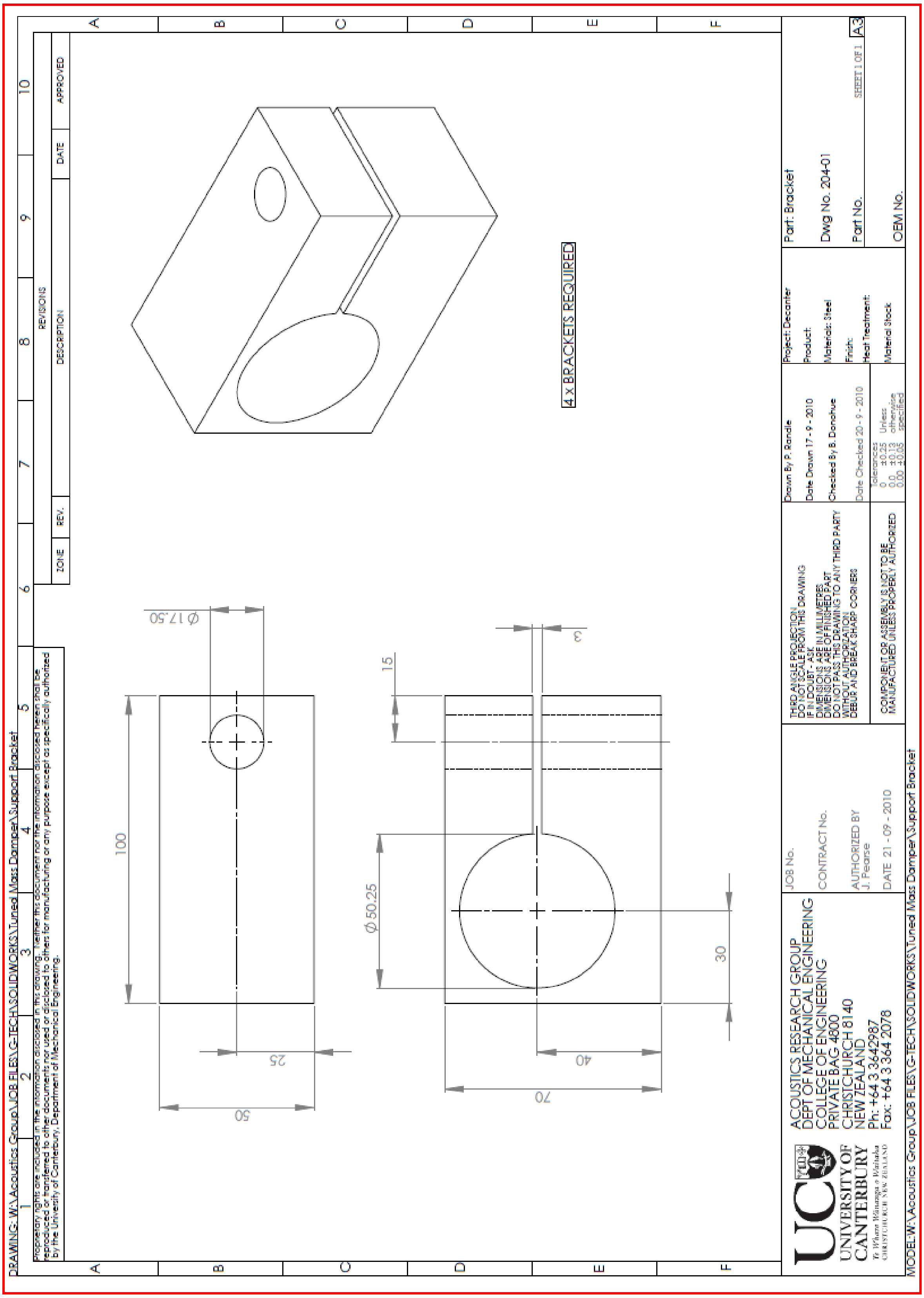
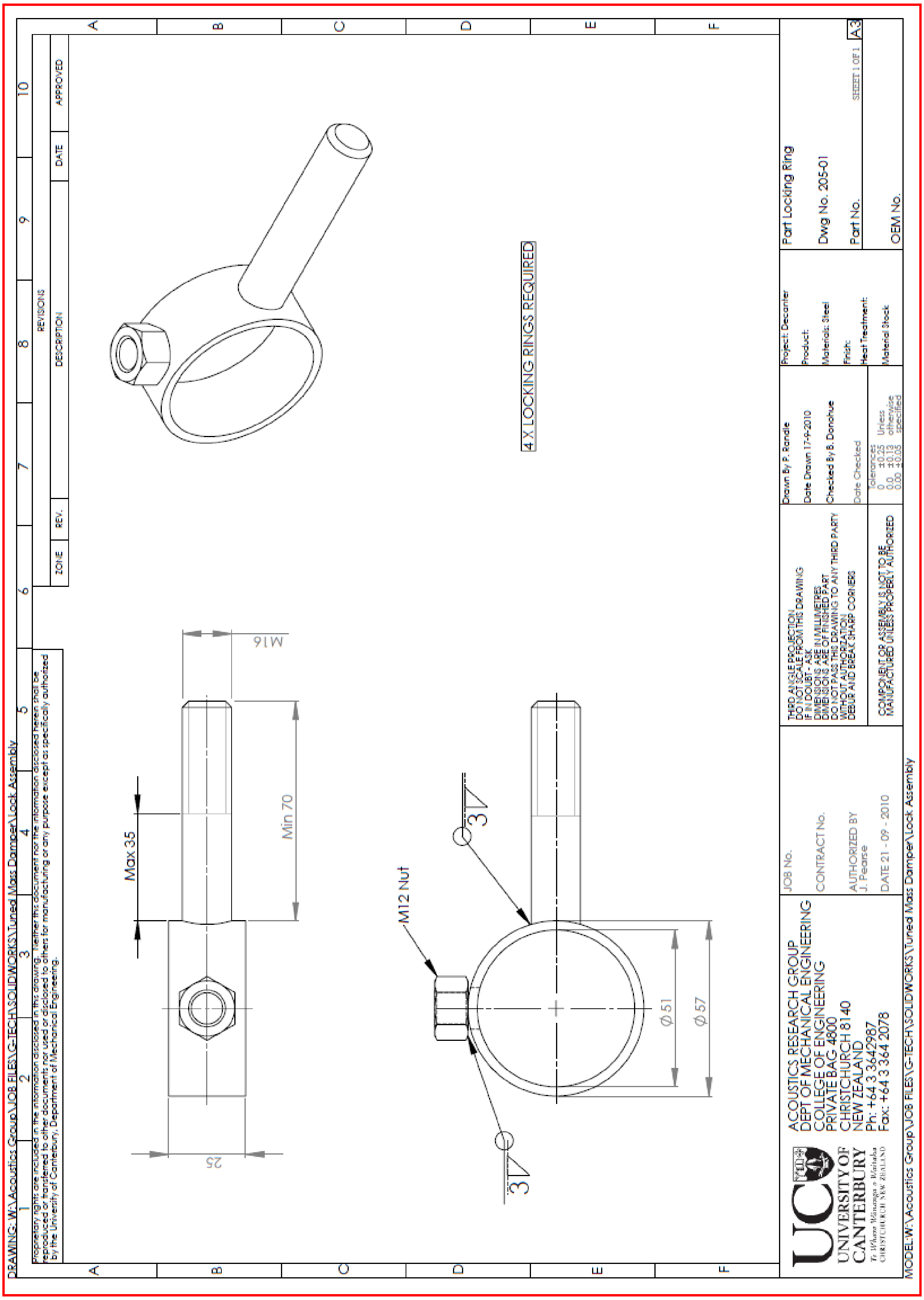
12 Appendix A – Tuned Mass Damper

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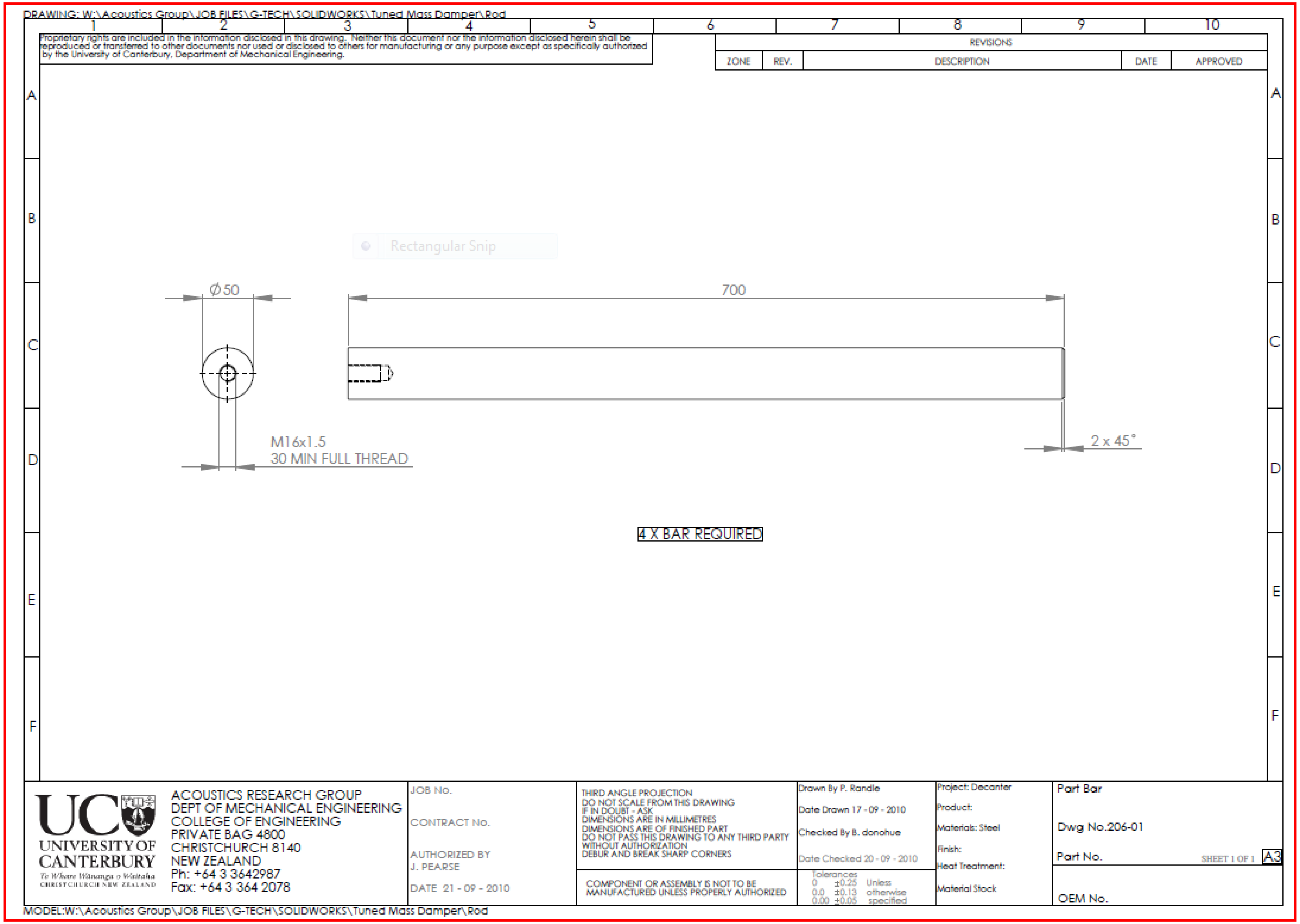

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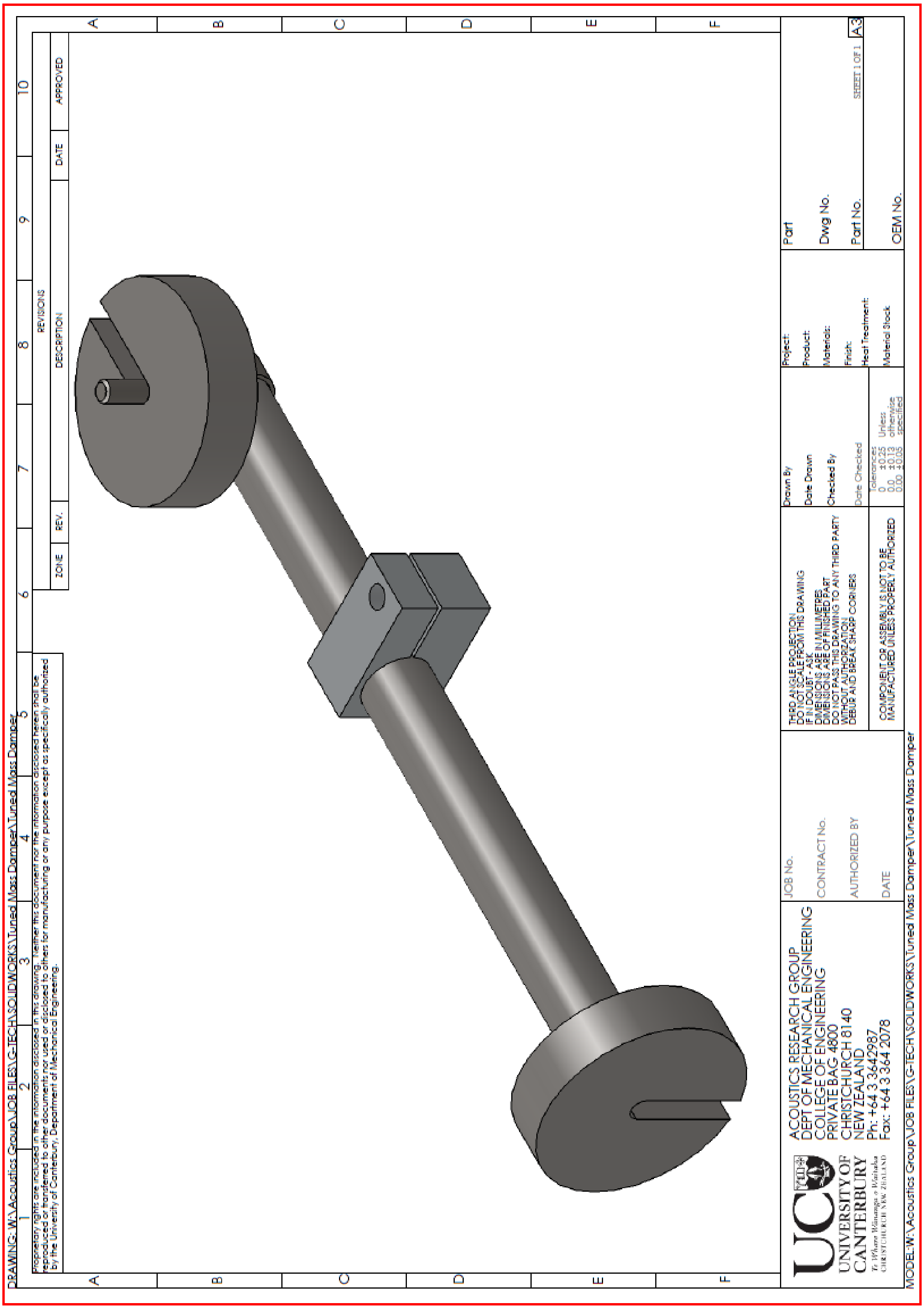
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